SPARK
REPORT

STRATEGIC PLANNING FOR APPLIED RESEARCH AND KNOWLEDGE

Science
Council of
British
Columbia

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REPORT OF THE NGV TASK FORCE

FOR: SCIENCE COUNCIL OF B.C.
ENERGY SECTOR PLANNING COMMITTEE

ABSTRACT: Environmental, economic and national energy security issues have resulted in a rapidly growing interest worldwide in natural gas for vehicles (NGV). British Columbia can capitalize on this interest by supporting a focused technological development strategy specifically targeted at marketable products and processes which can be developed and manufactured in B.C. This report describes general areas of required support, but focuses on these specific products for Science Council support: the non-metallic, on-board NGV storage cylinder; the direct injection natural gas fuel injection system for heavy-duty engines; the dual fuel diesel/natural gas retrofit system; and the small, on-site residential or small commercial NGV refueling compressor.

January 1989

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21 FEV of America, November 1988

The following quotation articulates the status of alternate fuel technology and the need for classical research and development in this field:

> "I believe it is worth emphasizing that we espouse the classical experimental design which precedes all successful transportation innovations, namely, fundamental research, controlled research and testing, limited field trials, then followed by larger (revenue-service) demonstrations. Unfortunately, the alternative energy scene is littered with attempts to short circuit this proven approach with frequently unfavorable results."*

1.0 MANDATE AND APPROACH

The Energy Sectoral Planning Committee Report of the B.C. Science Council (April 5, 1988) recommended that:

> "the B.C. Science Council recognize the strategic importance of research and development of natural gas fuelling of vehicles."

The Energy Sector Report further advised that "A management planning committee should be formed to identify the real opportunities and provide a rough schedule and cost estimate for implementing the specific initiatives."

Consistent with these recommendations, a Natural Gas Vehicle (NGV) Task Force was established at the October 27, 1988 meeting of the Energy Sector Planning Committee. The NGV Task Force consisted of the following individuals:

Bill Hennessey, B.C. Gas Inc. (Chairman)
Dr. Edward Auld, Physics Department, UBC
Dr. Phil Hill, Mechanical Engineering Department, UBC
Peter Neild, Carroll-Hatch (International) Ltd.
Michael Sinclair, Canadian Resourcecon Ltd.
Fred Feistmann, I.M.W. Compressors
Guff Muench, Cummins British Columbia
Dr. Malcolm Smith, B.C. Research

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* *R. R. Mayes, Transportation Perspective, Workshop on Alternate Fuels for Transportation: Canadian Research Needs, May 9-10, 1988, Ottawa.*

The NGV Task Force followed the same general Terms of Reference subsequently adopted by the Energy Sector Planning Committee on November 24, 1988. These included:

1. To implement the recommendations of the SPARK Energy Report.

2. Maintain communications and information flow between the Energy Sector activities and those activities in the other sectors that have an interest in the new developments in energy.

3. Maintain a coordination and cooperation role with other energy research initiatives.

4. Take a pro-active stance in recommending any research and development initiatives in energy that might provide an important economic impact to the province, and report these findings to the Steering Committee.

The primary focus of the NGV Task Force was seen as identifying specific areas of research and development in the field of NGV that could be undertaken in British Columbia in order to:

- develop, manufacture, and sell products to both the domestic, Canadian and international NGV marketplaces, providing income and employment growth in the province;

- facilitate the displacement of refined petroleum products made from out-of-province crude oil with a clean-burning domestic fuel, natural gas. This has obvious advantages in terms of provincial energy self-sufficiency, keeping income and jobs in the province and improving air quality.

In identifying areas of NGV research and development that should be undertaken in British Columbia, the Task Force assessed potential R&D efforts in terms of the following criteria:

1. **Does the R&D Program result in the development of a product within a reasonably short time-frame?**

   Projects should have well-defined and realistic objectives, and be targeted at a the development of a specific product or process, as opposed to research merely for the sake of increased knowledge.

2. **Will there be a market for the product resulting from the R&D effort, either in B. C. or elsewhere?**

   In order to be marketable, the product must encourage the increased use of NGV through cost reductions, superior efficiency, increased reliability, performance and safety, reduced emissions levels, lighter, safer, or more compact fuel storage, greater convenience in use, or through technical developments which allow natural gas to be used in new vehicular applications. Also, will the market still exist by the time the product is developed, or will it not exist a few years from now due to changes in technical, economic, or regulatory circumstances?

3. **Does B. C. have any comparative advantages relative to other regions to justify undertaking the R&D initiative?**

   Such comparative advantage could consist of a body of professional expertise such as the extensive work that has been undertaken in B. C. in the field of dual fuel natural gas/diesel engine conversion system development. The work should not duplicate work which is already underway elsewhere.

   Alternatively, certain types of NGV R&D might be justified in B. C. because a unique or major market exists in this province (e.g. NGV use in logging and mining trucks). Certain types of capital-intensive activities (e.g. dedicated heavy-duty spark-ignition NGV engine manufacture from the ground up, as opposed to local assembly) may not be viable to undertake in B. C., as opposed to other regions of Canada and the U.S. where specialized infrastructure in machinery and equipment industries already exists.

4. **What is the potential of the product or process for employment creation and provincial value-added?**

Activities which are capital-intensive but not labour-intensive, not only tend to generate less employment and provincial value-added, but are also inherently more risky in the event of a market downturn.

Bearing these criteria in mind, the Task Force undertook a review of the status of the NGV market and NGV technology on a worldwide basis in order to identify those areas where R&D efforts would yield dividends to British Columbia through the development, manufacture and sale of products to the domestic and international NGV marketplaces.

2.0 THE MARKET

There is an existing market for natural gas use in light and medium duty vehicles (less than 15,000 kg GVW) in Italy, Canada, United States, New Zealand, Argentina, France and Russia, and developing interest in Indonesia, Malaysia, Mainland China, Venezuela, and numerous small, generally Third World countries with proven natural gas reserves (Appendix 1).

There also appears to be a broad effort worldwide to develop the technology to allow the use of natural gas in vehicles presently powered by diesel engines. The primary driving force for NG to replace diesel fuel in the United States and Canada is to reduce air pollution, while other countries focus on economic benefits, into which is factored energy self-sufficiency and balance-of-trade.

Besides natural gas, the only other fuel being popularly considered as a large scale alternate to petroleum fuels is methanol. Methanol is a technically interesting fuel but is a value-added natural gas or coal product requiring energy-intensive production in capital-intensive plant, and has questionable emission benefits (Appendix 2). Propane is also a clean burning alternate fuel. However because it is produced as a by-product of oil refining or is stripped from natural gas, propane supplies are limited and supply is tied to the extraction rate of natural gas and processing rate of crude oil, usually a few percent of throughput. Also, many third world countries prefer to allocate propane supplies to the residential market to conserve trees.

3.0 TECHNOLOGY

NGV technological development can be summarized worldwide as follows, where the countries of origin are listed roughly in order of market share:

3.1 NGV Conversion Equipment for Gasoline Vehicles

(a) Established Technology

NGV Carburetion Equipment - Italy, United States, Holland
Electrical Ancillaries - Italy, United States, Holland, New Zealand
NGV Storage Cylinders - Steel: Italy
- Metal/Composite: Italy, USA

On-board High Pressure Gas Components (lines, fittings, valves) - New Zealand, Italy, USA, Japan

(b) Developing Technology

Electronic engine control systems (for improved power and fuel economy and decreased emissions) - Canada, United States (both CGA funded) and Japan

Composite Cylinders - Sweden, France, Switzerland, Canada

Adsorption Gas Storage - United States, Canada

3.2 Diesel Fuel Replacement (Vehicular Applications)

Dual fuel (natural gas with pilot diesel fuel ignition) - Canada, United States

Single fuel spark ignition NG - New Zealand, Australia, Canada, United States

3.3 Large Fueling Station Technology

Established technology - Canada, New Zealand, United States

Advanced technology - Canada (developing)

3.4 Small Compressor Technology for Residential and Small Commercial Applications

No established product yet, but development is ongoing in Switzerland, New Zealand, Italy, United States and Canada.

4.0 OPPORTUNITIES FOR BRITISH COLUMBIA

Canada is the only country worldwide with a broadly based NGV research and development program, although significant investments are being made in other countries for development of specific NGV products and for demonstration projects.

Canada's NGV research is being coordinated and funded through the Canadian Gas Association (refer to project list, Appendix 3). Contributors to the fund include natural gas utilities, pipeline companies, the federal government and one or two provinces. British Columbia firms have received a disproportionately large percentage of CGA Research and Development Funds to date, but provincial government leverage funding and other factors are resulting in an eastward movement of new projects.

The spin-off of basic and applied research is often the development of products and processes. For this reason, the Task Force is concerned that the Province of British Columbia is not a contributor to the CGA NGV Research and Development Fund. Provincial participation together with Science Council guidance would permit selective leveraged support of CGA projects which would provide the best job creation potential for British Columbians.

The NGV Task Force has evaluated the status of the NGV market and technology worldwide and has identified the following areas where opportunity for new product development exists and where expertise and interest has been demonstrated in B. C.

4.1 Non-Metallic Cylinders

The high weight of steel cylinders presently used for NGV storage acts as a significant impediment to NGV use, not only in cars and light trucks, but also in heavy trucks where the use of heavy tanks could reduce payload, and therefore revenue.

Recently, the aerospace and defense industries have demonstrated the viability of fibre-reinforced plastic (FRP) pressure vessels on applications up to 10,000 psi (NGV storage is at 3000 psi working pressure).

The development of a domestic composite NGV cylinder industry could significantly reduce the weight penalty and would also result in substantial import duty and transportation cost savings relative to the present situation, in which nearly all cylinders are sourced from Italy. The local availability of cylinders would also eliminate the need to carry extensive cylinder inventories, as the turnaround time between the ordering and receipt of cylinders would be greatly reduced.

In addition to the NGV industry, composite cylinders would be sold to a number of other markets, including aerospace and defense industries.

Gas adsorption technology is technically interesting, but extensive research and investment by others has not yet come close to the storage density of pure compressed gas.

A technical paper on cylinder technology is attached (Appendix 4).

Recommendation

A consortium of three British Columbia companies, combining research and development, manufacturing and marketing expertise, has expressed interest in manufacturing all-composite NGV cylinders in B.C. for the world market. The objective of this project is rated by the Task Force as very deserving of Science Council support. Justification is as follows:

- All-composite cylinder design is not restricted by limitations of diameter (within broad limits) or length. They therefore find universal application on the smallest car and the largest truck. Deep drawn steel cylinders are limited in both diameter and length.
- The physical plant required to manufacture full composite cylinders is modest and can be expanded in increments as required to meet sales volumes.
- The technology is advanced and proven in concept, but the production and marketing of these cylinders specifically for NG applications has not yet occurred.

The cost to develop the designs of full composite cylinders for the world market would be about $800,000 and would require one year for development and testing to field demonstration stage. Full production could begin in 1990.

There may also be design requirements for cylinder safety systems; that is devices to protect against high temperature and pressure. A further $100,000 could be required for this area of research.

4.2 Heavy Duty Natural Gas Engine

A large capital investment is required by a vehicle owner to use natural gas. The vehicle classes most economically suitable for natural gas are therefore the largest fuel users such as railway locomotives, mine haulage trucks, buses and Class 6 and larger trucks (19,500 lb GVW and up).

There are a number of fueling options to use natural gas in internal combustion engine (see figure in Appendix 5). These options may be broken into two broad categories - homogeneous and heterogeneous charge internal combustion, describing the condition of the air and fuel inside the combustion chamber, essentially at time of ignition. A general description of each and a review of research to date in each area, is as follows:

Homogeneous mixtures occur with premixed fueling systems such as carburetion, throttle-body injection and port injection, but excluding pre-stratified charge port injection. In low turbulence combustion chambers, combustion proceeds at the flame speed of natural gas. Ignition is by diesel pilot, glow plug or spark plug.

In high turbulence combustion chambers (swirl and/or squish induced turbulence), combustion proceeds at a much faster rate (Appendix 6).

Low turbulence combustion characteristics:

- requires less energy from engine
- susceptible to detonation from compression ratios (CR) as low as 12.7:1 (engine specific) (Appendix 7)
- high charge air temperatures
- susceptible to lean misfire

High turbulence combustion characteristics:

- swirl reduces volumetric efficiency
- squish requires power from the engine
- less susceptible to detonation (Corvette/Lotus Appendix 8)
- less susceptible to lean misfire

High turbulence chamber design can dramatically increase the allowable compression ratio and result in better fuel economy. For instance, the Lotus-designed Corvette engine has an 11.25:1 compression ratio, yet can run on 87 octane gasoline (Appendix 8).

Heterogeneous mixtures can be produced by pre-stratified charge (PSC) (Appendix 9) or by direct injection.

South West Research is developing a PSC system for the Cummins L-10 engine on contract from Southern California Gas and Columbia Gas, of Ohio. The engine will be used on buses manufactured by Flexible, of Delaware, Ohio.

Direct injection is accomplished in a similar manner to diesel fuel injection. Southwest Research (Appendix 10), Sulzer (Appendix 11), and Wartsilavasa (Appendix 12) have reported results on direct injection of natural gas, and Wallace and Green (Appendix 13) have developed a direct injection system for hydrogen.

The major advantages of direct injection are:

- potentially lower NO_x due to longer-duration release of heat energy and cooler temperature due to lean mixtures;
- less knock potential;
- less intrusive to classical proven diesel engine design;
- closer to proven, pure diesel cycle;
- comparable fuel efficiency to straight diesel operation.

Disadvantages are:

- must use LNG or develop a natural gas vapor compression technique to maintain about 3000 psi injection pressure. While ships have ample room for an on-board gas compressor, space on trucks and buses is very limited. The challenge therefore is to develop a compact compression system for use on vehicles.

4.2.1 Ignition Systems

Whether homogeneous or heterogeneous mixtures are used, natural gas will not reliably ignite at typical diesel engine temperatures and pressures (Appendix 11). Ignition assistance is required. Ignition systems considered for natural gas are:

(a) Diesel Pilot

Tests on many engines show that from five to fifty percent pilot fuel is required for reliable, detonation-free ignition of natural gas at full load. Engines with no more than pilot diesel fuel capability are called pilot fuel-ignited natural gas engines. Engines with capabilities to operate on straight diesel and on pilot-ignited natural gas are called dual fuel diesel engines.

(b) Spark Ignition

Spark-ignited engines are true Otto Cycle engines with combustion taking place in the same manner as the common automobile gasoline engine, or if used on direct injected natural gas engines, may be called modified diesel cycle engines.

(c) Glow Plug Ignition

Methanol engines have operated using glow plugs, but early reports indicate poor reliability. Spark ignition, or fuel cetane improvers are normally used on methanol engines as a substitute for glow plugs (Appendix 14).

(d) Plasma Ignition

This has been studied for many years and laboratory models have been tested, but no commercially acceptable devices are on the market (Appendix 15).

4.2.2 Comparison of Technologies

B.C. Research has studied dual fuel engine characteristics (Appendix 16). Conversion to dual fuel is less intrusive to original equipment manufacturers (OEM) engine system so dual fuel diesel/NG may be easier to integrate into the market in the short-term. OEM warranty concerns, marginal economics (at present) and uncertain emission benefits may not make this an attractive option for the long-term. Dual fuel, however, is very attractive as a market-builder for dedicated natural gas use in heavy duty applications because vehicle range is not limited by on-board natural gas storage capacity or the availability of NGV refueling stations.

Spark ignition (SI) is mechanically and electrically straightforward, but the high compression ratios and high energy release of heavy-duty engines place large thermal and mechanical stresses on the spark plugs. The plugs must also operate at higher discharge energy levels than for gasoline engines, further stressing the plugs and high tension delivery system. However, spark plug ignition is successfully used by electric utilities such as B.C. Hydro which operates spark ignited Cooper Bessemer engines/generators at up to 15:1 compression ratios and three megawatt electrical output. Transport Fuel Systems, of New Zealand and Hamilton St. Railway (Ontario) all report good spark plug life on heavy-duty (formerly) diesel engines (Appendix 17).

Conversion of an existing diesel engine to spark ignition is costly, but relative costs to "build up" a spark ignition engine based on diesel engine components are not known.

Exhaust valve wear is higher on natural gas (Appendix 18) but methods exist to reduce the wear to almost the same as for diesel engines.

Diesel cycle efficiencies are generally higher than for the Otto cycle because of the lower (knock-limited) compression ratios and throttling losses of the Otto cycle. However, Raine and Elder (Appendix 17) report that an SI natural gas engine at 15:1 CR in bus service achieved equal cycle efficiency (±5%) to the equivalent diesel engine. Noise levels were also lower. Lower noise levels are a widely reported benefit of natural gas engine operation.

Goetz et al (Appendix 19) report about an 18% lower cycle efficiency on spark ignition natural gas, 12:1 CR, compared to the equivalent diesel engine (relative to diesel N = 100%). Indicated thermal efficiencies (ITE) of up to 49% have been reported using natural gas (Appendix 7).

Adiabatics Inc. (Appendix 20) reports that brake specific thermal efficiency up to 43.6% is possible with an advanced spark ignition natural gas (SING) engine. Regarding emissions, Goetz reports higher HC and NO_x, lower CO and particulates on SING.

In the future, diesel engines will become cleaner burning, primarily through control of the combustion process and improvements to the quality of diesel fuel. Diesel engine exhaust gas after-treatment has not yet been perfected, and the technical hurdles to develop such a system are formidable.

For SING engines, EGR and exhaust catalysts are proposed methods of reducing emissions, but the Task Force is not aware of the extent of possible emission reductions with these technologies. FEV of America (Appendix 21) reports that significant reductions in natural gas exhaust emissions are possible, and the proposed methods employ existing proven technology.

Fiba Canning claims to have met the 1991 Bus emission standard and the 1994 truck engine standard with a high compression GM 454 engine with a three-way catalyst (Appendix 21). Exhaust gas emission reduction is the area most in need of research for natural gas to be a serious contender in the North American engine market.

The potential emissions reduction capability of natural gas use in vehicles, together with its low cost and plentiful supply in many areas of the world is the primary justification for investment in research and development in the natural gas vehicle industry.

Summary - Heavy Duty Natural Gas Engines

Dual fuel diesel/NG and spark ignition NG have been used on heavy-duty stationary engines for many years. Engine life increases of 100% on NG are possible in this application. Cycle efficiencies can be close to diesel engine efficiency, noise can decrease, much lower emissions are possible, but emission reduction systems must be optimized to achieve this potential.

Converting existing engines to dual fuel diesel/NG is easier and less costly than converting to SING, but driver optional use of natural gas makes dual fuel less credible from an emissions perspective. Dual fuel diesel/NG however has the major advantage of permitting a gradual development of the truck market for natural gas. The range of a dual-fuel truck is not limited by the on-board natural gas storage or by the availability of NGV refueling stations.

Pilot-fuel only ignition eliminates some of the concerns about dual fuel, but the continued use of some diesel fuel may preclude exhaust gas after-treatment. Meeting future emissions standards with diesel-pilot fuel natural gas may not be possible. Pilot fuel only vehicles will also suffer the same range limitations as single fuel vehicles, and development in the early stages of market will be limited to captive vehicles such as buses and mine-haul trucks.

Direct injection of natural gas may have superior emission and efficiency benefits compared to homogeneous mixture systems. Laboratory testing should be undertaken to quantify the effects of direct injection.

Plasma and glow plug ignition systems may increase efficiency and reduce emissions, but such systems have not yet proven to be reliable.

Spark ignition system technology is very advanced, and with careful heat shielding and plug selection, has proven reasonably reliable on high compression SI engines.

Pre-stratified port injection systems may have the benefit of reduced NO_x emissions, but the design of such systems for on-road applications will be very challenging.

Recommendation

The Task Force believes that natural gas is an excellent fuel for both captive vehicles (buses, local pickup and delivery vehicles) and for the general truck population. The best fueling option for captive vehicles is single fuel natural gas, and for general trucking, is dual fuel diesel/NG. As NG refueling stations become more prevalent, single fuel natural gas for almost all applications will be preferred. Details are as follows:

(a) Single Fuel

The ultimate method of fueling an engine with natural gas appears to be by direct injection with some form of ignition assistance. This method appears to be the only fueling option which provides the same power and efficiency of the high output diesel engine without risk of detonation. Ironically the major design challenge is not the direct injector, but a device to compress the natural gas to the required injection pressure.

Proposals for designing direct injection NG systems (including gas compression) have been made by a B. C. University and a B. C. Consultant. The best potential for job creation in B. C. is to encourage and foster the development of retrofit direct injection natural gas systems for existing diesel engines such as are supplied by Wartsilavasa (Finland) for their marine engines. We believe that a truck engine manufacturer would cooperate in such a venture.

Single fuel direct injection natural gas systems development particulars:

- 3-year program;
- $1,500,000 approximate cost;

- possible participants: UBC (proposed for joint Science Council/Gas Research Institute funding), UVIC, B.C. Research.

This program could carry the technology to the Research Laboratory Prototype stage.

(b) Dual Fuel

Dual fuel diesel/natural gas is considered to be an excellent market-builder for natural gas in the truck market, primarily because vehicle range is not limited by on-board natural gas storage or availability of NGV refueling stations.

B. C. organizations have already demonstrated expertise in developing dual fuel natural gas systems. An existing dual fuel design proposal for Science Council funding deserves immediate attention.

Dual fuel diesel/natural gas systems:

- 1-year initial program (one engine type);
- $400,000 approximate cost;
- possible participants: B.C. Research, Cummins B.C. (proposed for Science Council funding).

4.3 Small Refueling Systems

Small refueling systems for home or commercial use are required to reduce the inconvenience of frequent refueling as required with natural gas. Home refueling may also reduce vehicle conversion cost, weight and volume of the conversion system, because on-board storage for only one day of driving is required.

On-site natural gas refueling for commercial vehicles will likely reduce the labour cost for time spent going to and from the public station and time spent actually refueling vehicles.

BC Gas is aware of five small compressor design projects worldwide. In fact, BC Gas has evaluated most of the prototype units available. All are suitable for fuelling only one or two vehicles daily usually on an overnight fill. A range of compressor sizes is required to fuel as little as one car and as many as 20 trucks. One small compressor, designed by Sulzer of Switzerland, has reached the market demonstration stage.

For development of small compressor to prototype stage in British Columbia, approximately $500,000 and one to two years is required.

Recommendation

With the very large market potential for a small home or commercial natural gas compressor, the Science Council should consider support of initiatives from British Columbia companies for development in this area.

4.4 Large Refueling Systems

Presently all high capacity quick-fill NGV refueling systems worldwide are built around classical designs of reciprocating compressors. Thus, refueling systems consider compression and storage as two separate functions.

Other concepts such as hydraulic accumulators and recompression of partially depleted storage could dramatically decrease filling time, increase storage utilization and reduce cost.

Recommendation

Although projects of this type have not been proposed to Science Council, favorable consideration should be given to future proposals in this area. A budget of $500,000 and one year duration would be required for product development. Much of this investment ($250,000) could be recouped by sale of the final prototype to a user.

5.0 SUMMARY

Canadian, and in particular, British Columbia companies have shown considerable initiative and imagination in commencing projects to design and manufacture products for the rapidly developing world NGV market. However, British Columbians are competing with others in Canada and elsewhere who are well funded by their provincial governments and/or by large corporations.

For British Columbia to maintain a leadership role in the NGV industry and to capitalize on the very significant job creation potential of NGV product manufacturing, a focused technological development strategy is required. This report details specific target products, the rational for selection, a proposed budget and time-frame for each.

6.0 RECOMMENDATIONS

The industrial revolution began an acceleration in the rate of change of knowledge. Recently, computers and telecommunications have not only dramatically increased the rate of change of knowledge, but they have also broadened the availability of knowledge to all areas of the world. For British Columbians to compete in the information age, we cannot depend on osmosis, or on individual effort to foresee opportunity and to act on that opportunity. We must instead establish a permanent watch-dog body of experts, each transcending different areas of knowledge and expertise. The main body of experts would interact with sub-groups with specific expertise to identify and define opportunities for British Columbia. The B. C. Government could then act as a catalyst for private industry development of new technology.

The first recommendation from the NGV Task Force is as follows:

(a) The Government of B. C. through the Science Council of B. C. should perpetuate the SPARK process. The Energy Sectoral Planning Committee (ESPC) should continue to foresee and to recommend action on energy opportunities. A NGV Industrial Opportunities Coordinator should be appointed to act upon the recommendations of the Science Council and to provide a communications link between Science Council, the CGA, Gas Research Institute (Chicago), the B. C. NGV industry and the B. C. Government. The thrust for the coordination of B. C.'s NGV development should come from industry; however the participation of the B. C. Government is critical to the long-term success of this initiative. The Coordinator should assess the potential for support on the same criteria as used by the NGV Task Force in Section 1.0 of this report.

Specific recommendations are as follows:

(b) Encourage, assist and financially support the development of:

- (i) fibre-reinforced plastic on-board NGV cylinders;
- (ii) direct injection retrofit natural gas fueling systems for heavy-duty diesel engines;
- (iii) dual fuel diesel/natural gas retrofit systems for heavy-duty diesel engines;
- (iv) small compressors for home or small commercial fleet use;
- (v) the British Columbia Government through Science Council of B. C. should provide co-funding to the Canadian Gas Association for research and development projects destined to be undertaken in B. C.

An example of the need for a permanent watch-dog body and improved communication with government is demonstrated from recommendation (b)(i). Inland Natural Gas and B. C. Hydro Research and Development have a development proposal for FRP NGV cylinders before Science Council. The proposal is languishing while other suitors are trying to move the project to Alberta, Ontario or the U.S.

The NGV Task Force welcomes this opportunity to provide recommendations to Science Council on this subject.

The process has been rewarding to the participants and illuminating as to the opportunities available to the people of B.C. through the continued use of the Sectoral committees and the SPARK process.

APPENDICES 1 - 6

1 International Gas Union

2 Butane - Propane News, November 1988

3 CGA Annual Report

4 Advanced On-board Storage of Natural Gas

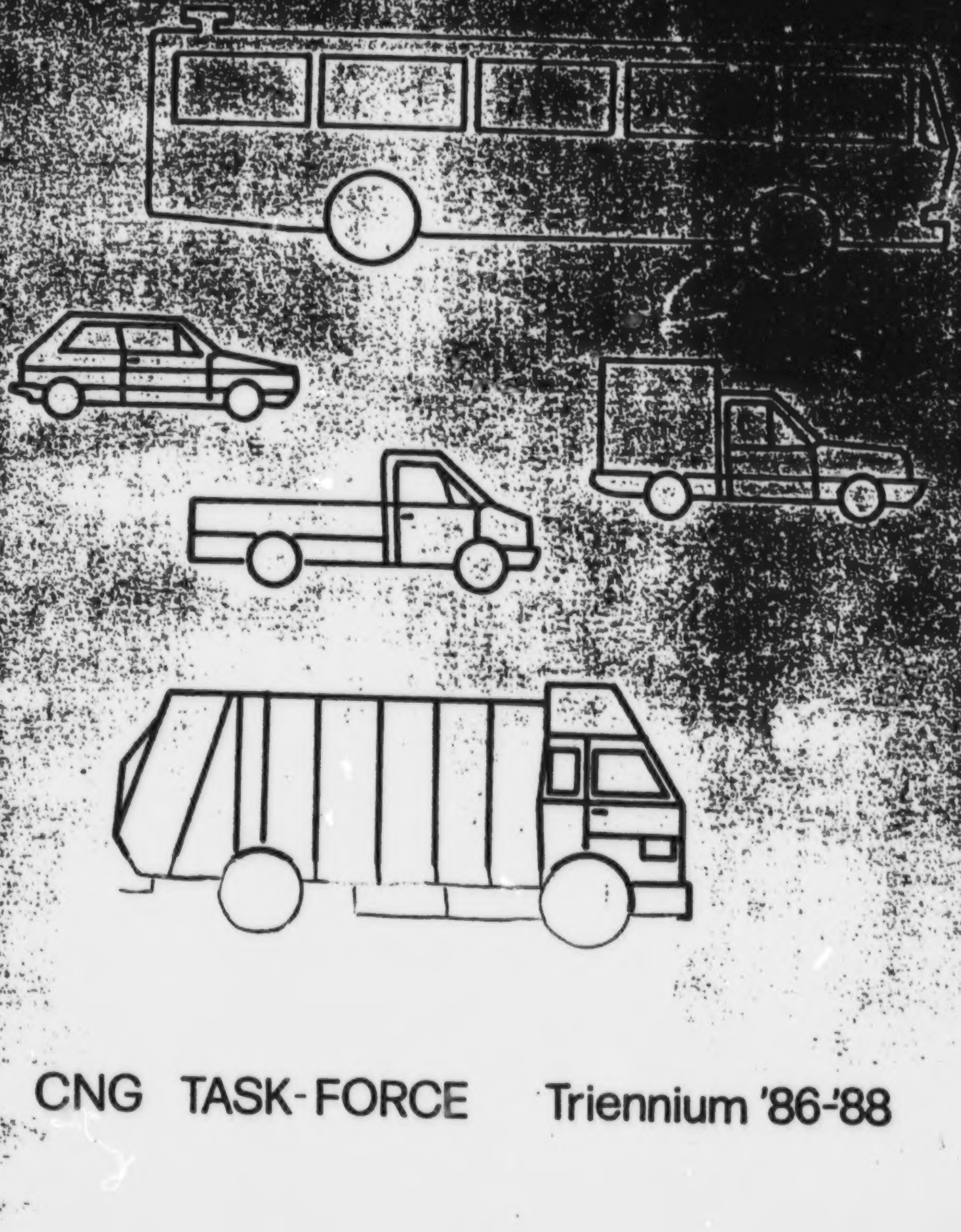
5 Fueling Options

6 Fast Burn Engine Development for Spark Ignition Engines

APPENDIX 1

INTERNATIONAL GAS UNION

NATURAL GAS FOR VEHICLES: A GLOBAL PERSPECTIVE



CNG TASK-FORCE Triennium '86-'88

"NATURAL GAS for VEHICLES - a global perspective"

The long-term issues which caused such concern to heads of governments during the world oil crises of the 1970's have not disappeared. Security of supply through energy use diversification; environmental protection; balancing of energy reserves and the buffering of world economics from oil price shocks are all issues which remain with the world community today and are unquestionably destined to become more important strategic issues in the future.

Currently, no other technological innovation within the gas industry offers as much favourable potential to address these issues as the successful development of the NGV market in the transportation sector. In addition, the use of indigenous resources, namely natural gas, provides many countries with opportunities to reduce the unfavourable balance of payment accounts whith offshore exporting countries which supply the bulk of the world's liquid fuel requirements.

The substitution of natural gas for gasoline and diesel fuel does present significant economic and environmental advantages, particularly in the long term. Scientists have been concerned for many years with the emissions generated from gasoline and diesel fueled vehicles. the gradual conversion of gasoline and diesel vehicles to natural gas will undoubtedly reduce the incidence of respiratory disease in major urban centres with high populations of vehicular traffic.

It is generally conceded that natural gas is readily available in all of the developed countries of the world and is fast becoming more available and gaining market acceptance in the lesser developed areas.

The existence of natural gas distribution systems and national grids already in place provide the framework for developing a natural gas refueling infrastructure critical to market growth of the NGV market sector.

However, to date and on a global scale, little has been done to develop the market for NGV, notwithstanding the social, economic and strategic advantages of the product.

TOTAL PASSENGER VEHICLES (CARS)
CLASSIFIED BY FUEL USE
(Responding Countries Only)

Country	Total N of cars 000's	Diesel 000's	Gasoline 000's	LPG 000's	Natural Gas Vehicles	** Refueling stations
Argentina	3,773	N/A	N/A	N/A	10,000	27
Australia	6,300	55	6,245	30	100	20
Austria	2,530	N/A	N/A	N/A	NIL	NIL
Belgium	3,343	570	2,700	67	1	1
Canada	11,118	125	10,958	30	5,300	95
Columbia	829	N/A	N/A	N/A	30	1
Czechoslovakia	2,575	N/A	N/A	N/A	30	N/A
Denmark	1,500	N/A	N/A	N/A	11	4
France	20,600	N/A	N/A	N/A	5,000	N/A
Finland	1,609	136	1,467	N/A	NIL	NIL
Indonesia	746	N/A	N/A	N/A	500	N/A
Iran	5,000	N/A	N/A	N/A	1,250	1
Ireland	709	32	673	4	23	1
Italy	20,888	1,726	18,216	716	229,626	230
Japan	23,939	N/A	N/A	N/A	NIL	NIL
Netherlands	4,949	412	3,972	564	75	2
New Zealand *	1,000	N/A	890	N/A	110,000	399
Pakistan	372	N/A	N/A	N/A	250	N/A
Spain	9,500	650	9,300	12	NIL	NIL
Sweden	2,730	99	2,641	1	NIL	NIL
Switzerland	2,678	43	2,635	NIL	NIL	NIL
United Kingdom	16,827	N/A	N/A	N/A	NIL	NIL
U.S.A.	130,364	N/A	N/A	N/A	30,000	255
U.S.S.R.	10,500	N/A	N/A	N/A	N/A	180
Total	284,379	3,848	59,697	1,424	392,196	1,216

* North Island
Numbers are rounded

** Total number of refueling stations (public, private, experimental).

Fig. 19

"NATURAL GAS for VEHICLES - a global perspective"

The long-term issues which caused such concern to heads of governments during the world oil crises of the 1970's have not disappeared. Security of supply through energy use diversification; environmental protection; balancing of energy reserves and the buffering of world economics from oil price shocks are all issues which remain with the world community today and are unquestionably destined to become more important strategic issues in the future.

Currently, no other technological innovation within the gas industry offers as much favourable potential to address these issues as the successful development of the NGV market in the transportation sector. In addition, the use of indigenous resources, namely natural gas, provides many countries with opportunities to reduce the unfavourable balance of payment accounts whith offshore exporting countries which supply the bulk of the world's liquid fuel requirements.

The substitution of natural gas for gasoline and diesel fuel does present significant economic and environmental advantages, particularly in the long term. Scientists have been concerned for many years with the emissions generated from gasoline and diesel fueled vehicles. the gradual conversion of gasoline and diesel vehicles to natural gas will undoubtedly reduce the incidence of respiratory disease in major urban centres with high populations of vehicular traffic.

It is generally conceded that natural gas is readily available in all of the developed countries of the world and is fast becoming more available and gaining market acceptance in the lesser developed areas.

The existence of natural gas distribution systems and national grids already in place provide the framework for developing a natural gas refueling infrastructure critical to market growth of the NGV market sector.

However, to date and on a global scale, little has been done to develop the market for NGV, notwithstanding the social, economic and strategic advantages of the product.

APPENDIX 2

WOGA Spokesman Dings Methanol Before California Air Resources Board

Chevron USA's Bruce Beyaert points out the many shortcomings of the controversial fuel.

Long the darling of the quick-fix politicians (along with gasohol), methanol has only recently come under the scrutiny of concerned scientists. Of chief concern are unresolved health issues, technical problems, the questionable safety of methanol, and its cost.

Speaking before the California Air Resources Board (CARB), Bruce Beyaert, representing the Western Oil and Gas Association (WOGA), addressed the problems presented by methanol as a motor fuel.

Beyaert, a business planner with Chevron USA's home office in San Francisco, pointed out that WOGA's member companies are in the energy business and that, regardless of the fuels required, most of those companies would continue to be the prime suppliers of motor fuels. He said that many of the companies have large natural gas resources around the world which could provide transportation fuels in the form of either compressed natural gas (CNG) or methanol. The companies' service stations could be modified to sell methanol and/or CNG, and five of the member companies have methanol production capacity. Two of the companies (Arco and Chevron) are already cooperating with the California Energy Commission (CEC) to help refuel the current fleet of some 600 methanol-fueled vehicles. LP-gas, Beyaert said, is another product which WOGA members already produce and retail that can be used as a motor fuel.

Alcohol-Fueled Vehicles

Beyaert said that vehicles can operate on gasoline-alcohol blends (gasohol) but such fuels create photochemical smog, rendering them useless in such areas as Los Angeles and Denver.

Ethanol is used as a motor fuel in Brazil, but it is much more expensive than methanol and offers less, if any, potential to improve air quality. Beyaert said that studies have shown that large-scale conversion of corn to ethanol would drive up the price of foods such as milk, eggs, chicken, and beef through increases in the prices of agricultural feedstocks.

Although methanol has long term potential as a vehicle fuel, it also has major problems that must be resolved.

Beyaert said that based on very limited photochemical modeling work done in California, methanol spark-ignition vehicles have the potential to slightly reduce peak ozone concentration *if* formaldehyde exhaust emissions can be controlled to the same level as gasoline vehicles. However, Ford and General Motors say that they don't know how to control formaldehyde emissions from methanol vehicles to the same low level achievable with gasoline vehicles.

Formaldehyde Emissions

Testing by Ford, GM, and Chevron research has consistently shown that current generation, multi-fuel methanol vehicles emit 30-40 mg/mile of formaldehyde when running on methanol at low mileage. This is three to five times more than from the same vehicles operating on gasoline.

Data for these multi-fuel vehicles are available only up to about 15,000 miles of service, Beyaert said, but formaldehyde emissions from methanol are expected to increase steadily as mileage accumulates. Formaldehyde emissions from older 1983 Ford Escorts, which operate only on methanol, are from 40 to well over 100 mg/mile at 25,000 to 40,000 miles—much higher than that expected from comparable gasoline engines.

In addition to being a very "hot" photochemical reactant, formaldehyde is toxic in its own right. EPA has listed it as a "probable human carcinogen" and has completed a risk assessment on it. Formaldehyde is also undergoing a risk assessment by the California Department of Health Services as a "toxic air contaminant."

Beyaert said it is essential to evaluate the highest formaldehyde concentrations that may occur. Summer air-shed model studies are not adequate. Peak formaldehyde emissions are higher due to the longer warmup time required before vehicle catalysts become effective. He also said that poor ventilation and high exposure situations should also be examined: for example, tunnel, parking garage, and urban street canyon situations and high exposure occupations such as bridge toll collectors, and parking garage workers.

"We recognize that gasoline is hardly a benign substance and that benzene is a leukemogen," said Beyaert. "The point is that it would be poor public policy to stimulate widespread use of a new substance without understanding the consequences." Even under summer conditions, and without undemonstrated, excellent formaldehyde control assumed, studies show that formaldehyde concentrations in downtown Los Angeles may be up to about six times greater than the highest benzene concentrations measured anywhere in the South Coast Air Basin in 1986; and the average formaldehyde concentration substantially exceeds the average benzene concentration. Even higher formaldehyde concentrations would be likely in the winter.

Methanol Toxicity

Beyaert said there is serious concern about control of exposure to prevent poisoning of motorists and mechanics because of the high toxicity of methanol. Ingestion of less than one ounce (in siphoning fuel for instance) can cause permanent blindness. Methanol is also readily absorbed by the skin. This means poisoning, even blindness, may result from misuse of the fuel as a solvent or hand wash, or from spilling methanol on a pant leg or skirt when refueling a car.

Although methanol's acute toxicity is well understood, said Beyaert, there is a dearth of information about the effect of long-term, chronic exposure to low concentrations in the atmosphere. Another unresolved issue is the so-called "cancer fog" effect. This involves the potential formation of secondary toxic pollutants such as dimethyl sulfate from water-phase reaction of methanol in fog or clouds.

Methanol's mild taste and odor would make it difficult to tell when ground water has been contaminated. Also, a report by Cambridge Analytical Associates indicates that leakage of a tank containing methanol would contaminate a six-fold greater area at a concentration 100 to 1000 times higher than occurs when a gasoline tank leaks. The report also warns that the contaminated groundwater would turn anaerobic and generate methane, creating a flammability hazard over a wide area.

Methanol Safety

Pure methanol (M100) burns with an invisible flame and could cause extensive personal injury. Even where M85 (85% methanol and 15% gasoline) is used as a fuel, M100 will have to be stored and blended somewhere in the fuel distribution system, creating a need for additional safeguards and a greater awareness by those involved with the fuel.

Bulk storage of methanol has its problems also, said Beyaert. For example, the vapor space above M100 in a storage tank will normally be in the flammable range at temperatures between about 40° and 120°F. Even with M85, the vapor space above the liquid will normally be in the flammable range at temperatures up to +10°F. The vapor space above gasoline is not flammable at these storage temperatures. Because of the flammability problem, alcohols are typically stored either in tanks having floating covers or an inert atmosphere. According to Beyaert, safe storage has not been addressed satisfactorily, nor has the safe transportation of the fuel in barges, marine vessels, and tank trucks.

Beyaert said that the existence of flammable methanol vapor concentrations in bulk storage tanks, underground tanks in service stations, vapor recovery systems, and in the fuel tanks of motor vehicles is of major concern to the fuels industry and to fire departments. Ignition of vapor by any means could rupture the tank or system and fully engulf it in flame. He said that control of the product in bulk storage plants should not be a major problem to solve. No easy solutions are yet apparent for dangers that exist in vapor recovery systems, service station tanks, or motor vehicle tanks. Past experience has shown, Beyaert said, that ignition of the vapor space can occur during repair activities or motor vehicle accidents involving dispenser pumps and vapor recovery piping.

The burning of methanol also adds to the "greenhouse effect." Carbon dioxide emissions from the methanol fuel cycle are higher than those created when using gasoline from crude oil. The increase is about 10% if the ethanol is made from natural gas, but it increases to about 80% if the methanol comes from coal.

The current price of methanol is high. Mile for mile, methanol costs about twice as much as gasoline, and three times more than diesel fuel.

Current global methanol manufacturing and shipping capacity is minuscule when compared to even California's transportation fuel markets alone. "Ninety new world-scale methanol plants would be required to replace California's current consumption of gasoline and highway diesel fuel," Beyaert said.

Such an effort would require new tankers and a capital investment ranging from $40 to $70 billion, based on current technology, California port sizes, and individual 2500-metric ton-per-day methanol plants. (These estimates do not include the costs of developing additional gas fields and pipelines to provide feedstock and fuel for the methanol plants.)

Additional investments would be required for large new port facilities in California capable of handling methanol tankers, Beyaert said. The volume of imports would be 15 times the current marine imports of crude oil and petroleum products. Also, the California offshore fuel distribution and marketing system would have to be doubled in size because methanol contains about half as much energy as gasoline and diesel fuel. This means that pipeline capacity, terminal and bulk plant storage, tank trucks, service station storage, and pumping capacity would all have to be doubled in size.

The Chicken or the Egg?

Beyaert said that the private sector could undertake such a massive investment program only if investors believe that methanol vehicles and fuel will receive wide public acceptance. This means that the vehicles must perform satisfactorily in all respects, the fuel cost must be competitive, and the public health consequences, including safety, must be understood. Private sector investors do not want to set up a new fuel system without every assurance that a viable market exists.

Speaking to the proposal that California government might mandate methanol fuels, Beyaert argued that this would work only with solid consumer and voter acceptance. Even with those factors, a mandate could backfire and be rescinded promptly if unsuspected or inadequately investigated health and safety issues led to excessive disease, deaths, or environmental impacts. "What government invokes it can revoke," said Beyaert.

Cost-Effectiveness

Crude oil prices would have to increase dramatically before methanol could compete with low-emission gasoline vehicles. When all the costs of operating methanol vehicles are weighed against per-ton or emission reduction for reactive organic gases (about $400,000 per ton for methanol vehicles), it may be much less expensive to design low-emission gasoline vehicles.

Virtually all of the $40-$70 billion investment required for methanol plants and tankers would occur in foreign countries. The plants would be built where natural gas reserves substantially exceed local market needs and where there are markets that can be economically reached by pipeline. Methanol plants are not likely to be built in the U.S. due to the outlook for higher natural gas prices, which would make them noncompetitive. U.S. methanol imports are already mushrooming to meet chemical demands.

While WOGA member companies do have large natural gas reserves in locations such as Canada and Australia, most of the surplus natural gas is located in the USSR and OPEC countries, and they have the capability to dominate global natural gas prices and methanol trade.

Beyaert pointed out that reliance on foreign methanol would require a massive outflow of funds from the U.S. First, methanol costs twice as much as gasoline at current crude oil prices and, second, it is more expensive to import a finished transportation fuel.

While iterating that most WOGA member companies have the potential to supply a wide variety of transportation fuels, including methanol, Beyaert said that no fuel with potential for practical, widespread use is absolutely benign or completely "clean." His recommendation to CARB was that the best way to reduce emissions from motor vehicles is to set stringent emission standards and enforce them. "This will allow vehicle manufacturers and fuel suppliers to use their creative talents to find the least-cost solutions which meet air quality needs consistent with consumer acceptance." ■

APPENDIX 3

ACCOUNTS THAT WERE OPEN IN THE 1989 FISCAL YEAR

Account Number and Descriptive Title		Contractor	<-------	Expenses Incurred	------->	Accumulated Holdbacks
			before 1989	in 1989	Total	
1. 1	Diesel Baseline Studies	B.C. Research	$85,977.00	$22.58	$85,999.58	$8,600.00
1. 3	Fast Burn Development for Spark Ignition Engines	University Of British Col	$31,509.00	$66,137.66	$97,646.66	
1. 8. 1	Map the Exhaust Gas on a Cummins L-10 fitted with GFI	Ortech International		$12,788.89	$12,788.89	
2. 2. 1	Dual Fuel Diesel/Natural Gas Engine Development	B.C. Research	$149,344.94	$53,972.19	$203,317.13	$20,396.60
2. 3	Oxygen Sensor Installation	CNG Fuel Systems	$27,508.00	$743.04	$28,251.04	
2. 4	NGV Electronic Fuel Injection Development - AirSensors	AirSensors Inc.	$233,659.08	$31,611.61	$265,270.69	$26,527.13
2. 5	Optimisation of GFI Cummins L-10 NGV Engine for Low NOx	Ortech International		$124,837.86	$124,837.86	
2. 7	Development of a Programmable Ignition Timer	Mogas Sales Inc.	$24,881.30	$4,632.53	$29,513.83	
2. 8. 1	Development of a Gaseous Fuel Injection System - ORTECH	Ortech International	$498,645.82	$697,698.61	$1,196,344.43	
2. 8. 2	GFI Patent Application Costs	Rogers, Bereskin & Parr		$36,970.61	$36,970.61	
3. 2. 1	Optimised NGV Refuelling Station Design	B.C. Gas Inc.	$92,061.61		$92,061.61	$9,206.18
3. 3. 4	Advanced On-Board NGV Cylinder Development	Powertech Labs Inc.	$87,356.34	$369,629.19	$456,985.53	$45,198.51
3. 3. 5	Investigation of Vehicle Cylinder Safety	Powertech Labs Inc.		$11,322.92	$11,322.92	$1,132.30
3. 3. 6	Investigation of Ground Storage Cylinder Safety	Powertech Labs Inc.		$140,262.27	$140,262.27	$14,026.22
3. 4. 1	Development of a Self-Service Reuelling Device	Canadian Gas Association	$147,635.00	$8,651.00	$156,286.00	
4. 4	Public Refuelling Station Compressor Data Base	B.C. Research	$28,564.61	$6,322.33	$34,886.94	$3,488.81
4. 5. 1	Test of Spark Ignition Cummins L-10 Engine Durability	Ortech International	$55,149.99	$42,190.88	$97,340.87	$9,734.90
4. 5. 2	Test of Dual Fuel Catapillar 3406B Engine Durability	B.C. Research	$250.51	$1,096.17	$1,346.68	$134.66
4. 6. 1	NGV ASI Conversion Kit Evaluation (AirSensors-BCR)	B.C. Research	$16,324.25	$476.42	$16,800.67	$1,680.14
4. 6. 2	Dual Fuel Conversion Kit Evaluation (Dual Fuel -Cryogas)	B.C. Research	$44,344.54	$5,551.72	$49,896.26	$4,989.62
4. 7	Course Training & Teaching Material Survey	Centennial College		$10,000.00	$10,000.00	
5. 1	NGV Safety and Regulatory Liason and Testing	B.C. Research	$15,258.00	$1,424.44	$16,682.44	$1,368.48
5. 3	Survey of NGV Equipment Certification Status	Mr. M. Morissette	$2,278.00	$2,116.37	$4,394.37	
5. 4	Develop Stress Corrosion Cracking Criterion for Steel Cyls	Powertech Labs Inc.	$19,500.02	$5,728.19	$25,228.21	$2,522.84
5. 6	Safety Analysis of RRA for Natural Gas Vehicles	R.F. Webb Corporation Ltd		$21,600.00	$21,600.00	$2,160.00
5. 7	Develop Improved NGV Drive Away Release Device	B.C. Research		$4,414.19	$4,414.19	$441.42
5. 8. 1	Develop Tests for NGV Cylinder Safety Relief Devices	Powertech Labs Inc.		$26,513.78	$26,513.78	$2,651.38
6. 1. 1	NGV R & D Administration 1986 to 1989	Various	$320,214.00	$130,638.25	$450,852.25	
28 Accounts Active in 1989		Totals	$1,880,462.01	$1,817,353.70	$3,697,815.71	$154,259.19

ACTIVE PROJECTS

		Total Value of Project	Parallel Funding	Project Status	<--- CGA NGV Research & Development Fund ---> Budget	Contract	Equity	Co-Funding
1. 3	Fast Burn Development for Spark Ignition Engines	$120,000		ACT		$120,000	$120,000	
1. 5	Fast Lean Burn Cummins L-10 Engine Design (Pistons)	$868,810	$210,000	ACT		$418,810	$209,405	$209,405
1. 8. 1	Map the Exhaust Gas on a Cummins L-10 fitted with GFI	$18,230		ACT		$18,230	$18,230	
2. 5	Optimisation of GFI Cummins L-10 NGV Engine for Low NOx	$585,699	$345,699	ACT		$240,000		$240,000
2. 8. 1	Development of a Gaseous Fuel Injection System - ORTECH	$1,229,344	$33,000	ACT		$1,196,344	$965,290	$231,054
2. 8. 2	GFI Patent Application Costs	$28,200		ACT		$28,200	$25,905	$3,105
3. 1. 1	Development of Large Volume Fast Refueling System	$41,027	$12,500	ACT		$28,527	$28,527	
3. 2. 1	Optimised NGV Refuelling Station Design	$235,010	$10,000	ACT		$225,010	$225,010	
3. 3. 4	Advanced On-Board NGV Cylinder Development	$716,086	$82,092	ACT		$633,994	$418,994	$215,000
3. 3. 5	Investigation of Vehicle Cylinder Safety	$445,000	$115,000	ACT		$330,000	$84,000	$246,000
3. 3. 6	Investigation of Ground Storage Cylinder Safety	$326,000		ACT		$326,000	$115,000	$241,000
3. 4. 1	Development of a Self-Service Refueling Device	$149,487		ACT		$149,487	$42,266	$107,221
3. 6	Self Service Fueling System Design Extension	$75,000		ACT	$10,000	$55,000	$65,000	
4. 5. 2	Test of Dual Fuel Caterpillar 3406B Engine Durability	$189,947	$90,000	ACT		$99,947	$99,947	
5. 6	Safety Analysis of RRA for Natural Gas Vehicles	$96,230	$20,230	ACT		$76,000	$76,000	
5. 7	Develop Improved NGV Drive Away Release Device	$54,000		ACT		$54,000	$54,000	
5. 8. 1	Develop Tests for NGV Cylinder Safety Relief Devices	$64,922	$15,000	ACT		$49,922	$49,922	
6. 1. 2	NGV R & D Administration 1990 to 1992	$550,000		ACT		$550,000	$550,000	
18 Projects	TOTALS	$5,792,992	$933,521		$10,000	$4,599,471	$3,147,496	$1,492,785

PROJECTS APPROVED BY PROJECT SELECTION COMMITTEE

		Total Value of Project	Parallel Funding	Project Status	<--- CGA NGV Research & Development Fund --->			
					Budget	Contract	Equity	Co-Funding
1. 7. 1	Assessment of the Environmental Impact of NGV - Definition	$15,000		APPV	$15,000			
1. 8. 2	Exhaust Gas After Treatment - Methane to Carbon Dioxide	$381,000	$200,000	APPV	$181,000			
1. 9	Lean Burn Oxygen Sensor Development	$50,600		APPV	$50,600			
2. 9	Development of Gas Regulator - Yugotech	$120,466	$87,566	APPV	$32,900			
2.10	Development of a NGV Fuelled Navistar DT-466 Engine				Proposal not accepted by Navistar			
2.11	Development of Cummins L-10 Head Optimised for NGV	$750,000	$500,000	APPV	$250,000			
3. 2. 2	Sonic Temperature & Pressure Compensator & Metering Valve	$40,000	$20,000	APPV	$20,000			
3. 3. 8	Aluminium Cylinder Sustained Load Crack Testing				For presentation to PSC January 1990			
3. 4. 2	Certification of a Self-Service Refuelling Device	$50,000	$30,000	APPV	$20,000			
3. 9	Development of a Slow Fill NGV Installation for Small Fleets	$111,000	$91,000	APPV	$20,000			
5. 8. 2	Development of Pressure Relief System for Large Cylinders	$50,000	$10,000	APPV	$40,000			
5.10	Assessment of Refueling Hose Durability				For presentation to PSC January 1990			
12 Projects	TOTALS	$1,568,066	$938,566		$629,500	$0	$0	$0

PROJECTS UNDER CONSIDERATION

		Total Value	Parallel	Project	<--- CGA NGV Research & Development Fund --->			
		of Project	Funding	Status	Budget	Contract	Equity	Co-Funding
1. 4	Investigation of NGV Combustion Characteristics	$400,000		PROP				
1. 6. 3	Improved Injection Development	$0		PROP				
1. 7. 2	Assessment of the Environmental Impact of NGV	$500,000		PROP				
1. 8. 4	Exhaust Gas After Treatment - Nitrogen Oxides and Methane	$0		PROP				
2. 2. 2	Dual Fuel Caterpillar 3406B - Project Extension	$85,000		PROP				
2.12	Introduction of Light Duty Mono-Fuelled OEM Vehicles	$0		PROP				
2.13	Introduction of Heavy Duty Mono-Fuelled OEM Vehicles	$0		PROP				
2.14	Investigation of Pre and Open Chamber Engine Designs	$400,000		PROP				
2.16	Development of a Universal Throttle/Governor Assembly	$250,000		PROP				
2.17	Demonstration of Ultra Low Emission Concepts of NGV	$650,000		PROP				
3. 1. 2	Support for TTC Refuelling Station Installation	$20,000		PROP				
3. 3. 9	Development of In-Situ AE Test Rig for Steel Cylinders	$35,000		PROP				
3. 3.10	Crashworthiness Testing of Full Composite Cylinders	$100,000		PROP				
3. 3.11	Investigation of Long Wrapped Cylinder Bending Moments	$60,000		PROP				
3. 5. 2	Sulthane Commercialization RRA RAMD Study	$80,000		PROP				
3. 7	Development of Public NGV Refuelling Station Design	$500,000		PROP				
3. 8	Development of Fleet NGV Refuelling Station Design	$400,000		PROP				
4. 6. 3	Additional Conversion Kit Testing	$380,000	$190,000	PROP				
4. 8	Determination of Engine Tolerance to Fuel Composition	$70,000	$20,000	PROP				
4. 9	Development of a Fuel Composition Sensor	$200,000		PROP				
4.10	Component Development and Testing	$300,000		PROP				
21 Projects	TOTALS	$4,430,000	$210,000					

COMPLETED PROJECTS

		Total Value	Parallel	Project	<--- CGA NGV Research & Development Fund --->			
		of Project	Funding	Status	Budget	Contract	Equity	Co-Funding
1. 1	Diesel Baseline Studies	$85,997		FIN		$85,997	$85,997	
1. 2	Natural Gas Fuelled Diesel Engine Technical Assessment	$76,937		FIN		$77,135	$76,937	
2. 1	First Generation Conversion Kit Upgrade	$322,981	$171,000	FIN		$151,981	$151,981	
2. 2. 1	Dual Fuel Diesel/Natural Gas Engine Development	$288,966	$850,000	FIN		$203,966	$203,966	
2. 3	Oxygen Sensor Installation	$28,251		FIN		$28,255	$28,251	
2. 4	NGV Electronic Fuel Injection Development - AirSensors	$280,864	$20,864	FIN		$260,409	$171,463	$88,946
2. 7	Development of a Programmable Ignition Timer	$29,514		FIN		$29,514	$29,514	
3. 5. 1	Evaluation of RRA Reliability & Durability	$45,905	$16,000	FIN		$29,905	$29,905	
4. 1	Small Fleet Compressor Evaluation (Saskatchewan Farm)	$239,250	$209,250	FIN		$30,000	$30,000	
4. 2. 1	New Zealand NGV Cost Investigation	$30,806	$15,000	FIN		$15,000	$15,806	
4. 3	Test of Chrysler 5.2L Engine Durability	$74,785	$9,500	FIN		$65,375	$65,375	
4. 4	Public Refuelling Station Compressor Data Base	$49,888	$15,000	FIN		$34,888	$34,888	
4. 5. 1	Test of Spark Ignition Cummins L-10 Engine Durability	$147,351	$50,000	FIN		$97,351	$97,351	
4. 6. 1	NGV ASI Conversion Kit Evaluation (AirSensors-BCR)	$36,500	$20,000	FIN		$16,500	$16,500	
4. 6. 2	Dual Fuel Conversion Kit Evaluation (Dual Fuel -Cryogas)	$94,250	$50,000	FIN		$49,341	$49,341	
4. 7	Course Training & Teaching Material Survey	$10,000		FIN		$10,000		$10,000
5. 1	NGV Safety and Regulatory Liaison and Testing	$25,258	$10,000	FIN		$16,673	$16,673	
5. 2	Investigation into Aluminium Burst Disk Safety	$78,000	$28,000	FIN		$50,000	$50,000	
5. 3	Survey of NGV Equipment Certification Status	$4,394		FIN		$4,394	$4,394	
5. 4	Develop Stress Corrosion Cracking Criterion for Steel Cyls	$38,170	$2,600	FIN		$35,570	$35,570	
6. 1. 1	NGV R & D Administration 1986 to 1989	$470,000		FIN		$470,000	$470,000	
6. 4	ORF Report on GFI Workplan	$47,000		FIN		$47,000	$47,000	
6. 5	ORF Installation of GFI Equipment	$105,000	$45,000	FIN		$60,000		$60,000
6. 6	ORF GFI Laboratory and Equipment Study	$176,295		FIN		$176,295		$176,159
24 Projects	TOTALS	$2,786,362	$1,512,214		$0	$2,045,549	$1,710,912	$335,105

PROJECTS BEING TRACKED FOR INFORMATION PURPOSES

		Total Value	Parallel	Project	<--- CGA NGV Research & Development Fund --->			
		of Project	Funding	Status	Budget	Contract	Equity	Co-Funding
1. 6. 1	Hi-Speed Injector Valve Develop (BCRC/ORTECH) EMR	$123,000	$123,000	INFO				
1. 6. 2	Direct Injection Development (UBC) BCRC	$182,452	$182,452	INFO				
1. 8. 3	Exhaust Gas After Treatment - Nitrogen Oxides (TC/EMR)	$1,050,000	$1,050,000	INFO				
3. 2. 3	Storage Pressure Optimization	$3,000	$3,000	INFO				
3. 3. 7	Investigation of Acoustic Emission Testing Techniques	$138,000	$138,000	INFO				
3.10	Enhanced Storage	$0		INFO				
4.11	Testing of Lubrication Oil for NGV Diesel Block Engines	$0		INFO				
5. 5	School Bus Crashworthiness Testing	$70,000	$70,000	INFO				
5. 9	BC Gas NGV Safety Investigations	$146,000	$146,000	INFO				
9 Projects	TOTALS	$1,712,452	$1,712,452					

PHASE I NGV R&D FUND ALLOCATION BY CONTRACTOR

COL #1 PROJECT SERIAL	COL #2 PROJECT TITLE	COL #3 COST TO NGV FUND	COL #4 CASH CONTRIBUTION	COL #5 IN-KIND CONTRIBUTOR	COL #4 CONTRIBUTOR	COL #5 CONTRIBUTOR	AMOUNT TO CONTRACTOR	TYPE OF CONTRACT**
1.1	DIESEL BASELINE STUDIES	$ 80000	NIL	NIL	Nil	Nil	UEC - $50,000; BCR - $30,000	COMP; JOINT
1.2	DIESEL TECHNICAL STUDY	$ 79495	NIL	NIL	Nil	Nil	SYPHER:MUELLER - $79,495	COMP
1.3	FAST BURN DEVELOPMENT	$ 120000	NIL	NIL *	Nil	Nil	UEC - $80,000; JVIC - $40,000	SS; JOINT
2.1	GEN I UPGRADE DEVELOP	$ 151981	NIL	$171600	ORTECH	CNG Fuel Systems	CNG FUEL SYSTEMS - $151,981	COMP
2.2	DUAL FUEL DIESEL/NG	$ 179866	$ 36000	$ 50000	Depends on field trial	Caterpillar	BCR - $178,366	COMP
2.3	OXYGEN SENSOR INSTALL	$ 29895	NIL	NIL	Nil	Nil	ORTECH - $29,895	SS
2.4	EFI NGV CONVERSION KIT	$ 171483	$ 80946	$ 20455	HYGAS and DME	Air Sensors	ASI - $208,427; BCR - $54,982	COMP; JOINT
2.6	LARGE DUAL FUEL ENGINE	NIL	$400000	$300000	TDC and DME	Bombardier and GMI	BOMBARDIER - $400,000	SS
2.7	PROGRAMMABLE IGNITION	$ 28000	NIL	NIL	Nil	Nil	MOGAS - $28,000	UP
2.8	EFI CORE/AUTO DEVELOP	$1025280	$171054	$111600	HYGAS and DME	ORTECH	ORTECH - $1,196,344	COMP
3.1	FAST REFUELING SYSTEM	$ 26000	NIL	$ 12500	Nil	Union Gas	UNION GAS - $26,000	COMP
3.2	REFUELING STATION	$ 225010	NIL	$ 10000 *	Nil	BC Gas	BCG - $52,510; IMW - $102,500; BCR - $60,000	COMP; JOINT
3.3	CYLINDER DESIGN/TEST	$ 387000	$253000	$ 20000 *	TDC; BC Gas	Various; UEC	BCH - $535,000; UBC - $80,000; BCR - $25,000	COMP; JOINT
3.4	SELF-SERVE REFUEL PROBE	$ 40413	$107221	NIL	EMR, DME and CGA members	Nil	CGA - $147,634	SS
3.6	BRA RAND STUDY	$ 29905	$ 16000	NIL	DME and CGA members	Nil	TES - $29,905	SS
4.1	SMALL COMPRESSOR EVAL	$ 30000	$209250	NIL	SeekAg and EMR	Nil	SRC - $239,250	SS
4.2	NZ COST INVESTIGATION	$ 15805	$ 15000	NIL	ConGas and Inland Gas	Nil	CONGAS - $10,805; INLAND - $5,000	COMP; JOINT
4.3	SI ENGINE DURABILITY	$ 65375	NIL	$ 3500	Nil	Chrysler Canada	ORTECH - $65375	COMP
4.4	COMPRESSOR DATA BASE	$ 34888	NIL	$ 15000	Nil	CGA members	BCR - $34,888	COMP
4.5.1	RAND TEST OTTO CYCLE	$ 97351	NIL	$ 50000	Nil	Cummins	ORTECH - $97,351	COMP
4.5.2	RAND TEST DUAL FUELING	$ 39947	NIL	$ 90000	Nil	Caterpillar	BCR - $39,947	COMP
4.6.1	KIT TEST AIR SENSORS	$ 16370	NIL	$ 20000	Nil	Air Sensors	BCR - $16,370	COMP
4.6.2	KIT TEST CRYOGAS ENG.	$ 44250	NIL	$ 50000	Nil	Cryogas Eng. Ltd.	BCR - $44,250	COMP
4.7	SVC SUPPORT AND TEACH	NIL	$ 10800	NIL	CGA NGV Committee	Nil	CENTENNIAL - $10,800	UP
5.1	SAFETY LIAISON	$ 14370	NIL	$ 10000	Nil	EMR	BCR - $14,370	COMP
5.2	AL CYLINDER PRD SAFETY	$ 50000	NIL	$ 28000	Nil	BC Hydro	BCH - $50,000	SS
5.3	NGV EQUIPMENT SURVEY	$ 2800	NIL	NIL	Nil	Nil	CONSULTANT - $2,800	SS
5.4	SSCC TESTING OF CYLINDER	$ 35570	NIL	$ 2800	Nil	CGA members	BCH - $35,570	SS
6.1	NGV R&D ADMINISTRATION	$ 470000	NIL	NIL	Nil	Nil	CGA - $470,000	SS
6.4	EFI DEVELOPMENT DEFINE	$ 47000	NIL	NIL	Nil	Nil	ORTECH - $47,000	SS
6.5	EFI EQUIPMENT PURCHASE	$:36	$178159	$ 65000	Some CGA members	ORTECH	ORTECH - $119,890; WOODS GORDON - $56,405	SS; JOINT
6.6	EFI LAB INSTALL	NIL	$60000	$45000	DME	ORTECH	ORTECH - $60,000	SS
	TOTALS							
	% OF CONTRIBUTIONS/COST							

* This project is partially or completely frozen so that CGA costs may have to be adjusted.

** COMP = Competitive Contract Award; SS = Sole Source Contract Award; UP = Unsolicited Proposal

Projects 2.5 and 3.5 are not included because they have yet to be awarded and final cost sharing is undetermined.

PROJECT SERIAL	PROJECT TITLE	ESTIMATED BUDGET	PLAN START DATE	PLAN STOP DATE	TYPE OF PROJECT	AUTHORITY LOGIC	POTENTIAL CONTRACTOR + = AND, / = AND/OR
1	FUNDAMENTAL RESEARCH	$2,500,000					
1.3.2	FAST BURN DESIGNED CUMMINS L-10 NG	$500,000	JUL 89	DEC 91	COMP, JOINT	3	ORTECH/BCR + UBC + UVI
1.4	NGV COMBUSTION CHARACTERISTICS	$400,000	JUN 89	JUN 93	SS, JOINT	3	UBC + UCAL
1.5	DIRECT INJECTION OF NGV	$1,000,000	JUN 89	JUN 93	COMP, JOINT	3	SRC/UBC/ORTECH/CONCORD
1.6	ENVIRONMENTAL MODELLING	$300,000	JAN 89	JUL 89	SS	1	GM CANADA
1.7	EXHAUST GAS AFTER TREATMENT	$300,000	MAR 89	MAR 90	COMP, JOINT	1	ORTECH/BCR + UNKNOWN
2	ENGINE DEVELOPMENT	$1,212,000					
2.2.1	DUAL FUEL CAT ENGINE EXTENSION	$85,000	JAN 89	DEC 89	SS	1	BC RESEARCH
2.9	CONVERSION SPECIFICATION DEVELOP	$127,000	JAN 89	JAN 90	COMP	1	UNKNOWN
2.10	OEM MONOFUEL NGV DEVELOPMENT	N/K	MAR 90	JUN 93	COMP	4	UNKNOWN
2.11	OEM NGV HDD ENGINE	N/K	MAR 90	JUN 93	COMP	4	UNKNOWN
2.12	PHASE III LARGE ENGINE DEVELOPMENT	$1,000,000	MAR 90	DEC 92	COMP, JOINT	3	BOMBARDIER+OTHERS
3	FUEL SYSTEMS DEVELOPMENT	$2,380,000					
3.3.5	VEHICLE CYLINDER SAFETY	$420,000	JUL 89	DEC 92	SS	1	BC HYDRO R&D
3.3.6	GROUND STORAGE CYLINDER SAFETY	$300,000	JUL 89	DEC 92	SS	2	BC HYDRO R&D
3.3.7	LNG CYLINDER MATERIALS	$630,000	JUL 89	DEC 92	SS	3	LAVAL
3.5.1	RRA RAMD GEN II STUDY	$80,000	JUN 89	JUN 91	SS	2	TES LIMITED
3.6.1	BUS FUEL PROBE DESIGN WORK	$50,000	MAY 89	DEC 89	SS	4	SHEREX INDUSTRIES
3.7	PUBLIC STATION DEVELOPMENT	$500,000	JUL 90	JUL 92	COMP, JOINT	1	UNKNOWN
3.8	COMMERCIAL STATION DEVELOPMENT	$400,000	DEC 89	JUL 91	SS	3	SRC
4	TESTING AND EVALUATION	$380,000					
4.6.3	CONVERSION KIT TESTING	$380,000	DEC 89	DEC 91	COMP	1	UNKNOWN
5	SAFETY SYSTEMS	$290,000					
5.6	NGV SAFETY SYSTEM ANALYSIS	$110,000	MAR 89	OCT 89	COMP	1	UNKNOWN
5.7	DEVELOPMENT OF BREAKAWAY DEVICE	$30,000	DEC 89	JUN 90	COMP	4	UNKNOWN
5.8	SAFETY INCIDENT INVESTIGATION	$150,000	AUG 89	JUN 92	COMP	2	UNKNOWN
6.1.1	NGV R&D ADMINISTRATION EXTENSION	$550,000	JAN 89	JUN 93	SS	1	CGA
	TOTAL	$7,312,000					

APPENDIX 4

ADVANCED ON-BOARD STORAGE OF NATURAL GAS

by

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and

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ABSTRACT

Technology for the storage of compressed natural gas on board is reviewed. All steel cylinders and more recently aluminum alloy or steel liners hoop wrapped (circumferential) with glass fiber reinforced plastics, are being used for gas storage at 20.7 MPa (3,000 psi). Field service experience to-date suggests that those cylinders are suitable for NGV on board application. All composite cylinders having alternate layers of axial (polar) and hoop wrapping on a thermoplastic liner offer a new concept for light weight cylinders.

An R&D program sponsored jointly by the Canadian Gas Association and Transport Canada will examine the efficient and safe storage of natural gas on board vehicles. This two year $770 K contract to B.C. Hydro is part of the pooled government-industry R&D fund of $7 M towards improving the technology required to introduce NGV into the transportation market. Objectives of the on-board storage program are presented along with preliminary results.

KEY WORDS: Natural Gas for Vehicles (NGV), On-board cylinders, hoop-wrapped, polar-wrapped

INTRODUCTION

It is common practice to store vehicle fuel on board in the liquid state at ambient temperature either under atmospheric pressure (gasoline and diesel fuels) or under a moderate pressure to prevent rapid evaporation (propane). Natural Gas, however, has a critical temperature of -82.5°C for liquefaction under pressure. Natural Gas for Vehicles (NGV) is therefore generally carried on board as Compressed Natural Gas (CNG) at ambient temperature. This gaseous fuel must be kept under a high pressure to avoid taking up unduly large on-board space and to get a reasonable mileage out of the vehicle between refueling. The service pressure for on-board storage of NGV in Canada is 20.7 MPa (3000 psi).

Vehicle engines optimized for natural gas fuel are not yet commercially available but conversions of gasoline and diesel vehicles to NGV are becoming popular. The cost of on-board storage as a percentage of total NGV conversion cost (1) can range from 28 percent (one cylinder for a passenger car) to 90 percent (multi-cylinders on trucks). The on-board cylinders must satisfy two other requirements. The weight of the cylinder must be kept to a minimum and the cylinder must be safe and reliable. Clearly the design of cylinders requires effort to ensure future development of the NGV market.

The Canadian Gas Association has a three year pooled government-industy Research and Development Fund which has attracted a total of $7 million Canadian towards improving the technology required to introduce NGV into the transportation market. It is expected that this fund will be extended until 1992. One of the five areas being addressed through this program involves on-board cylinder development. A contract totalling $770,000 Canadian has been awarded to the Research & Development Division of British Columbia Hydro and Power Authority. This multi-disciplinary program has two main sub-contractors: the University of British Columbia and B.C. Research Corporation. The contractor has established liaison with provincial regulatory agencies and Transport Canada to expedite acceptance of cylinders. Contacts have been made with the Compressed Gas Association (CGA) of U.S.A.[*1], the National Bureau of Standards, and the Department of Transport (DOT). The work is being carried out in collaboration with a number of manufacturers of on-board NGV cylinders. Other interested parties that are pursuing similar work e.g. South West Research Institute in U.S.A. and the Department of Scientific

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[*1] The abbreviation CGA will be used in this paper to mean Compressed Gas Association only. Canadian Gas Association will not be abbreviated to avoid confusion.

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Energy Sources Technology
Conference and Exhibit (ETCE) Houston
January 22 - 25, 1989

and Industrial Research in New Zealand, have also been contacted in order to avoid redundancy in research effort.

The objectives of this paper are:

- To review the state-of-the-art of the NGV on-board cylinders.
- To present an overview of the approach used in the Canadian Gas Association research project to address issues of safety and reliability as they relate to Canadian NGV operating conditions.

TYPE OF CYLINDER

There are three types of NGV cylinders:
a) Steel cylinders,
b) Hoop-wrapped,
c) All Composite.

Salient features of each type are as follows:

a) Steel Cylinders

They constitute the majority of an estimated one million total in use all over the world (primarily in Italy and New Zealand). NGV cylinder design specifications permit a thinner and more highly stressed wall than ASME requirements, the former reduce the weight of the cylinder. Cylinder burst pressures are typically 2.5 times the service pressure. Cylinder geometry is also noteworthy. Although the best end design is hemispherical, NGV cylinder ends are flattened to conserve on board space. Over flattening results in stress concentration at the transition region and fatigue failure has been observed in laboratory tests (2).

NGV cylinders typically use AISI 4130X grade steel. Control of minor impurities, sulphur and phosphorus is considered important to maintain a high product toughness. Fabrication is generally done by deep drawing of sheet blanks with intermittent annealing. Then the open end is heated and spun to close it before an opening is bored through the closed neck. The cylinder is quenched and tempered to the desired hardness and bore hole threading follows. Quality control is undertaken at various stages. An important control consists in hydrostatic proof testing of each cylinder followed by a NDT, (typically ultrasonic) inspection.

Cylinder dimensions are typically 254 mm (10 in.) to 330 mm (13 in.) diameter x 889 - 1219 mm (35 in. to 48 in.) in length and up to 90 l internal water capacity. Wall thickness is 9.5 mm (3/8 in). A 90 l cylinder weighs about 77 Kg (170 lbs). Weight can be minimized by using thinner walled cylinders through increasing the steel strength by tempering at a lower temperature. However other design trade-offs, discussed later, limit the maximum allowable hardness of the material. Longitudinal cross-section of a typical steel cylinder is shown in Figure 1(a).

b) Hoop-Wrapped Cylinders

Figure 1(b) shows a typical hoop wrapped cylinder. These cylinders are lighter on a volumeric basis vis a vis all steel cylinders. The cylinders consist of two structural components: a metal liner shaped like the complete vessel; a fiber reinforced plastic wrapping applied to the central cylindrical portion. In the cylindrical portion, design load is shared by the liner and the wrapping, while the metallic end portions carry the entire design load. A NGV cylinder with a service pressure of 20.7 MPa (3,000 psi) would have a burst pressure of 25.9 MPa (3,750 psi) for the metallic liner, and 51.7 MPa (7,500 psi) after the hoop wrapping. Selective prestressing of the cylindrical portion of the gas cylinder is designed so that under normal operating pressures, including maximum service pressure, the fiberglass wrapping carries a greater fraction of the total load on the wall than the metallic liner. In a non-pressurized condition, cylinder metal liner (cylindrical portion) remains under compression, and the composite wrapping under tension. This feature prevents fatigue cracking in the cylindrical portion of the metal liner.

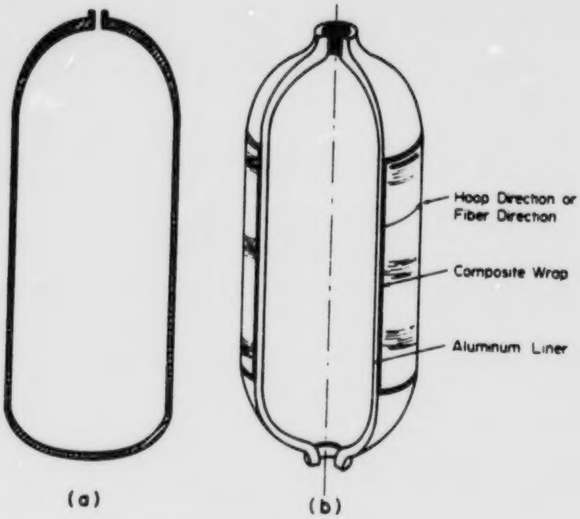

Fig. 1 Typical cylinder geometry
a) Cross section of a steel cylinder
b) A hoop-wrapped aluminum cylinder

Steel liners are made from quenched and tempered AISI 4130X type steels while aluminum alloys 6061-T6 and 6351-T6 have been use. The DOT FRP-2 standard, applicable to hoop wrapped cylinders, specifies only aluminum alloy liners and not hoop wrapped steel cylinders. Work in Canada is aimed at encouraging the use of the latter cylinders.

Steel liner fabrication consists of deep drawing from sheet stock. Aluminum alloys are extruded seamless tubes. Hot spinning is used for end closure of both metal liners. Hot die forging has also been used with aluminum. Since a relatively thin-walled steel material is involved, folds are likely to form as a result of die forging so it is not used for end closure of steel liners. Spinning for end closure can be conveniently carried out for cylinder lengths of less than 1829 mm (6 ft), end forging appears more suitable for longer cylinders. Following end closure and end-boring, liners are heat treated, threaded, hydrostatically proof tested and ultrasonically examined. Hoop-wrapping follows.

Continuous E-glass fibres in a matrix of polyester or epoxy resin are the most common hoop-wrappings.

Kevlar and graphite fibres have been used in other applications, but do not appear to be cost effective for NGV cylinders. The FRP (Fiber Reinforced Plastic) may be applied using a wet process in which case the E-glass roving is pulled through the liquid resin bath and wound over the side wall. Alternatively, a partially cured polyester E-glass tape may be used for winding. While the tape is generally more expensive it offers the option of tension wrapping to achieve the desired pre-stress. The resin is cured following wrapping. Roving is not amenable to pre-stressing hence wet wound cylinders generally undergo auto-frettage. The principle of auto-frettage is shown schematically in Figure 2.

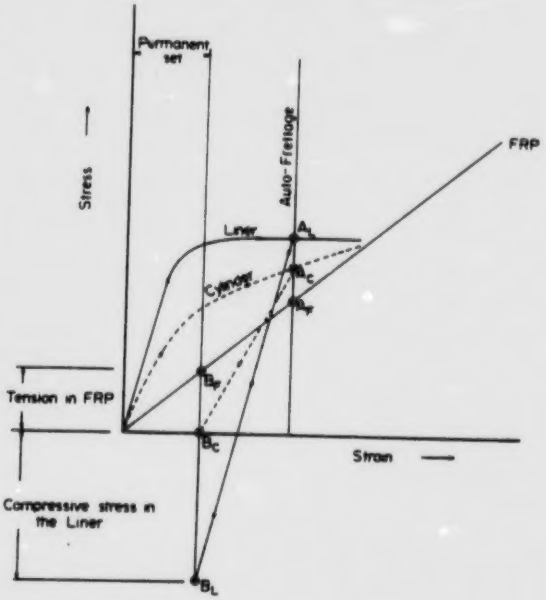

Fig. 2 Schematic illustration of the principle of auto-frettage. Letter A shows the stress-strain relationship upon application of auto-frettage pressure, Letter B after depressurization. Subscripts L, F and C refer to the liner, FRP wrapping and cylinder (hoop-wrapped) respectively.

The steps involved in auto-frettage are as follows. Hydrostatic pressure is applied to the cylinder until a pre-determined plastic deformation occurs (point A_L)at the cylindrical portion of the metallic liner (metallic end portions and the FRP wrapping remain in the elastic loading regime) then the cylinder is depressurized (point B). Thus the wrapping contracts and applies a compressive force on the liner (BL) but itself remains under tension (BF) due to the permanent set in the liner (BC). If the thickness and the yield strength of the liner material remain uniform over the entire cylindrical portion, then the FRP wrapping is uniformly stressed. Should there be localized variations in liner strength or thickness (e.g. due to microstructural variations) localized bulging or bending of long cylinders could be expected. Under such circumstances wrapping of the tape under tension offers a distinct advantage over auto-frettage. Autofrettage is commonly used for cylinders having a length of 1829 mm (6 ft) or less. Longer cylinders are tape wrapped under tension.

A 120 l capacity hoop-wrapped steel lined cylinder would be 330 mm (13 in.) in diameter, 1218 mm (48 in) long and weigh about 82.8 Kg (180 lbs). Hoop-wrapped aluminum alloy cylinders in sizes up to 8 m (26.2 ft) in length have been fabricated. The aluminum liner thickness is typically 12 mm (1/2 in.) and the wrapping 6 mm (1/4 in.). Sometimes a gel-coat is applied to the wrapping for UV and corrosion protection. In other instances fillers are added to the resin to minimize resin UV exposure.

c) All Composite Cylinders

These cylinders are very attractive due to their light weight. All composite high pressure cylinders have been designed and used for some time in military and aerospace applications. NGV applications are just beginning. One design of 50 l capacity and, 1829 mm (6 ft) in length may be used for NGV trucks. For a service pressure of 20.7 MPa (3,000 psi) these cylinders could have a 62 MPa (9,000 psi) burst pressure.

The production process starts out with either a polyethylene or cross-linked polyethylene liner moulded to the desired shape. Metallic inserts are placed on each end of the liner for threaded valve connections. An adhesive sealant attaches the metallic valve connections to the liner. E-glass rovings are pulled through a liquid resin and are wrapped on the polyethylene liner by a winding machine capable of controlling tension, angular rotation etc. After the wrapping process, the vessel is cured and hydrostatically proof tested. Continuous fibres are used for applying alternate layers of longitudinal and hoop wrapping. It is to be noted that DOT-FRP-1 Standard (revision 1, March 15, 1982) specifies resin impregnated continuous filament winding in both longitudinal and circumferential directions on a seamless aluminum liner as opposed to a thermoplastic liner.

SAFETY AND RELIABILITY CONCERNS

The DOT/CTC specifications were designed for the transporting of pressurized gases and results in thinner and more highly stressed cylinder walls than those of the ASME design. These increased stress levels require periodic cylinder retesting to ensure safety. Current inspection guidelines are those outlined by the Compressed Gas Association (3). These guidelines are based on structural stress analysis without sharp discontinuities. Sharp discontinuities, however do arise in service and lead to stress concentration. The standard deals with that situation by recognizing that at least on metallic cylinders sharp discontinuities on the exterior surface (such as localized pitting, crevice, line corrosion or even cracks) can be removed by grinding. Inspection is carried out following such conditioning. The reduced wall thickness (without sharp discontinuities) forms the basis for the rigorous fitness for purpose analysis. Acceptance or rejection of the vessel is thus based on the reduced thickness of the wall. Sharp defects on the interior surface of the cylinder are, however, not readily amenable to conditioning operations such as grinding. Of course non-destructive means are available to determine the depth of such internal surface cracks. The remaining wall thickness can be obtained. However, since the sharp defect is associated with stress concentration, the pressure retaining capability of the vessel containing the defect will be lower than that of a vessel having a wall thickness equal to the remaining ligament. Thus the standard requires hydrostatic cylinder testing to a pre-specified pressure (between service and burst pressure).

Currently, steel cylinders must be retested every 5 years, and hoop wrapped aluminum every 3 years, by hydrostatically loading to 1.67 times the service pressure. If permanent expansion exceeds total expansion by more than 10 percent, (5 percent for aluminum) the steel cylinder is rejected. More frequent retesting of the hoop-wrapped aluminum cylinders is considered necessary because of the uncertainties associated with a relatively new design and materials. The standard further imposes a service life restriction of 24 years for steel DOT-3HT cylinders. CGA-C6.2 on the other hand requires condemnation of all fiber reinforced high pressure cylinders 15 years after the date of manufacture. Neither the frequency of re-testing nor the longevity criterion are based on a rigorous fitness-for-purpose analysis. Safety concerns arise due to the following factors:

(a) Fatigue

Laboratory fatigue experiments on newly fabricated steel cylinders using cyclic hydrostatic loading between zero and service pressure suggest a fatigue life of approximately 10^5 cycles for the initiation and growth of cracks to cylinder rupture. The New Zealand Code requires the design to be proof tested on a prototype to 40,000 fatigue cycles without failure using a dynamic loading range between zero and service pressure (4). Although a cylinder may be designed for a service pressure of 20.7 MPa (3000 psi), in reality during the filling operation pressures as high as 24 MPa (3,500 psi) are permitted to compensate for the heating of the cylinder during refuelling and to maximize fuel storage. Fatigue life reduction by this process is unknown.

(b) Sulphide Stress Corrosion Cracking

Impurities, such as H_2S and CO_2, in the presence of water can create corrosion and stress corrosion cracking concerns. Aluminum alloys are considered less susceptible to such concerns than quenched and tempered steels. The degree of susceptibility of steel depends on the quality of the natural gas used and the hardness of the steel. Processed natural gas throughout Canada has remained of uniform quality over the past 35 years of major pipeline operations (5). In USA however, the composition of processed natural gas varies and there is no industry standard restricting contaminants in NGV fuel (6). Although the US gas transmission companies have the same gas quality standards as Canadian firms, there appears to be limited control in USA over gas sales from well heads. Consequently the DOT has severely limited the hardness of the steel to Rockwell hardness, R_c, of 22 in order to avoid sulphide stress corrosion cracking concerns.

Different steels can be compared using the standard NACE sulphide stress corrosion cracking test (7). This accelerated test, however, uses a far more corrosive solution than that encountered in NGV service. Consequently the NACE test data are not readily amenable to fitness for purpose assessment of a steel for a given environment. Recently a suitable modification of the NACE test method has been developed and used with steels having a range of hardness between R_c 18 and R_c 34 (8). This latter set of data suggest that steels with hardness numbers up to R_c 31 present no concern for sulphide stress corrosion cracking given the quality of NGV in Canada. Steel cylinder with hardness numbers as high as R_c 29 are currently permitted for use in Canada. Cylinders with hardness numbers as high as R_c 33 are being used in Europe and Australia. It is noteworthy, however, that the service pressure in Canada is somewhat higher than those permitted in Europe and Australia.

Higher strength/higher hardness number steels provide significant weight reductions. In quenched and tempered steels, however, the fracture toughness of the material is reduced as the hardness increases (i.e. the crack tolerance decreases as the strength increases). A balance must be struck in the design optimization process keeping the fitness-for-purpose in mind. Detailed fracture mechanics data are not yet available to carry out such an analysis. The South West Research Institute (SWRI) has a program underway to generate data on the R_c 22 material (9) which is of current interest in the US. The current Canadian Gas Association program will be generating similar data for R_c 30 material.

(c) Damage to the Exterior Surface of the Cylinder

As noted earlier the Compressed Gas Association guide-lines for inspection are based on pre-conditioning of sharp defects which consist generally of grinding out the defects. Dents on the cylinder wall are similarly considered as not significant stress raisers based on a critical ratio between the depth and major diameter of the dent. The question of pre-conditioning of sharp defects when applied to the fiber glass reinforced plastic wrapping of the hoop wrapped cylinders however, must be re-examined.

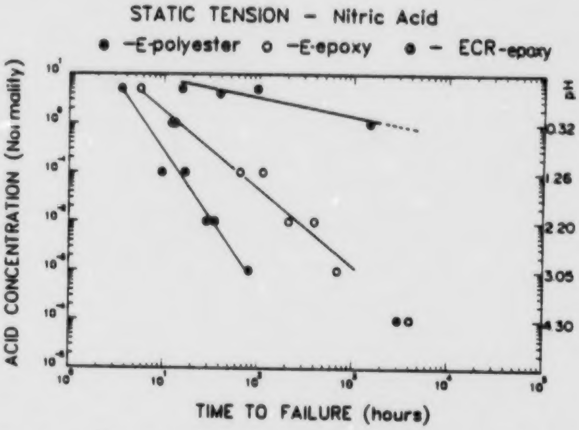

Fig. 3 Time to failure of linear FRP rods made with various glass-plastic materials. After reference 10.

E-glass fibers are used by most manufacturers of hoop wrapped and all composite NGV cylinders. E-glass is known to be more prone to stress corrosion cracking in a variety of environments than are other glass fibres available commercially (10). Figure 3 shows the relative performance of E-glass fiber composite rods with respect to ECR-glass, the latter is commercially available. Stress corrosion cracking of E-glass reinforced plastic on a hoop-wrapped cylinder was recently noted in the field. It occurred when battery acid was accidentally spilled over the cylinder. The cracks are shown in Figure 4.

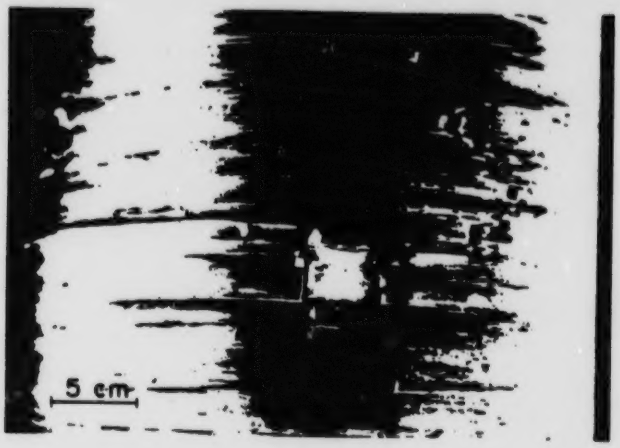

Fig. 4 Photograph showing stress corrosion cracks on a hoop-wrapped aluminum cylinder. Battery acid was accidentally spilled while the cylinder was in service.

(d) Sustained Load Cracking

The phenomenon of sustained load cracking has been recently studied in depth in the case of aluminum alloys (11). Alloy 6061-T6 appears to be free of this type of cracking. However, alloy 6351-T6 appears to be prone to this mode of cracking depending on minor impurities such as Pb, and grain size. Both alloys are being used for making hoop-wrapped aluminum cylinders. The phenomenon of sustained load cracking occurs at ambient temperatures under static load. Cracks have been observed (12) in the neck region of high pressure cylinders made with aluminum alloy 6351-T6.

(e) Vehicle Accident and Fire

In the event of a vehicle fire, will the cylinder pose a threat to life? To answer that question one must consider the material behaviour at elevated temperatures. Steels have a high melting point (≈1300 C) and undergo little creep deformation until temperatures as high as 500 C are reached and maintained for substantial lengths of time. Aluminum alloys on the other hand melt at about 600 C and hence substantial and rapid creep deformation can be expected at temperatures as low as 250 C.

Fiberglass composite resin tends to burn quickly in a fire. Even though the glass fibers may retain their ability to carry load to a higher temperature, will the gas leak out from an all composite cylinder? What would be the leakage mode? Does that present a hazardous situation? Is there a need for a fire resistant coating? Should there be a pressure-temperature relief device (PRD)? If so what would be the required characteristics of the PRD? These questions remain as yet unanswered.

THE CURRENT RESEARCH PROGRAM

The objectives are as follows:

- Re-qualification of existing cylinders in service.
- Development of a computer model for the Vehicle Fire
- Provide technical interface between manufacturers, utilities and regulatory bodies.
- Optimization of NGV cylinders.

(a) Requalification of existing cylinders

Hydrostatic re-testing necessitates removal of the cylinder from the vehicle, cleaning, immersion in a water jacket, and following the test, the cylinder must be dried before being replaced on board. The process is time consuming and expensive. An in-situ non-destructive test method would be desirable. The viability of in-situ acoustic emission testing through 110 percent (of service pressure) over-pressurization has already been demonstrated and accepted for steel trailer tubes for the transport of industrial gases. A special DOT approval has been obtained by Union Carbide for the acoustic emission test method in lieu of hydrostatic tests (13). The present study develops test methods for re-testing of 4130X steel cylinders with a hardness of R_c 30, and for hoop wrapped aluminum alloy cylinders. It is expected that the program will be extended to include steel hoop wrapped and all composite cylinders.

The fitness for purpose approach is based on the concepts of elasto-plastic fracture mechanics. Material properties such as crack growth, in the worst case environment, will be examined using cylinder material samples. Defects of known dimensions will then be introduced in the cylinder wall.

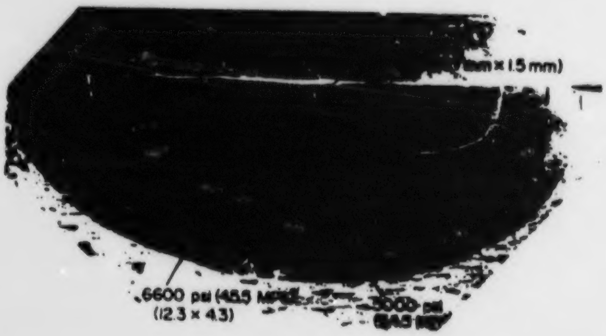

Fig. 5 Growth of a controlled crack introduced longitudinally on the exterior surface of the steel cylinder. Electric Discharge machining (EDM) was used for initiating the fatigue crack. Stable crack growth was monitored subsequently as the cylinder was hydrostatically pressurized. Acoustic emissions were also monitored.

Axially orientated cracks of various lengths, and semi-elliptical geometry (1:4 aspect ratio) are introduced by electric discharge machining followed by fatigue crack growth. An example of such a crack is shown in Figure 5. Cylinder surface fatigue crack growth in the axial direction is monitored using the Fractomat-Krak-Gage instrumentation; in the radial direction (depth) using an A-C potential drop method.

Stress analysis for that crack geometry is available in the literature (14). For the hoop-wrapped cylinders axial defects in the fibreglass hoop-wrapping only are being considered. Acoustic emission test procedures for fiber composites are already well established (15,16).

Crack growth is monitored on selected cylinders as a function of fatigue cycling from 2.4 - 24 MPa (350 - 3500 psi) to simulate the fuelling operation. In a second set of experiments a series of cylinders, each having a predetermined crack length, is subjected to hydrostatic loading while the acoustic emissions are monitored. The objective is to define the necessary threshold setting and correlate the acoustic emissions with the extent of cracking. This exercise will be followed by acoustic emissions while filling cracked NGV cylinders. The acoustic emission retest frequency and the associated rejection criteria will be defined through this effort for both steel and hoop-wrapped aluminum cylinders.

(b) Development of a Computer Model for the Vehicle Fire

Vehicle fire experiments are an expensive exercise. Fire testing with a pressurized fuel cylinder requires a remote site with adequate protection for test personnel. Such fire tests are necessary, however, to gain insight into the time, temperature and pressure rise relationships. In an attempt to simplify and standardize the test method, DOT have drafted a document (17) which involves making a controlled bon-fire. The cylinder is tested using that bon-fire. The bon-fire test is generally carried out inside a bunker. Such a facility is available at B.C. Hydro for the testing of the cylinder. For a development task related to the pressure release device (PRD), a simulation of the temperature and pressure increase with time has also been used in the laboratory (18).

Since vehicle fire characteristics appear to be fairly well known, field data could be used to model the temperature of the cylinder wall with time and heat transfer to the gas in the cylinder (and associated pressure rise). Making use of the material properties such as strength at various temperatures and associated creep rates, it should be possible to predict the time to rupture of the cylinder for pre-determined initial pressures. The computer model will be initially cross checked for its predictions against the fire test data available to date. The validated model could then be used as a design tool to obtain the following information:

- Is a PRD needed? If so, what should be its characteristics? What should be its location along the cylinder? How many are needed? Can some other NGV component suffice?

- Alternatively, can the cylinder be protected in the event of a fire using a thermal barrier coating? If so, what should be the characteristics of the coating? What are the critical locations?

A combination of the two approaches may provide optimum protection in some designs.

c) Provide technical interface between manufacturers, utilities and regulatory bodies

New technologies often result in products which do not satisfy existing codes. One approach used for the introduction of such a product into the market place is to carry out an engineering fitness-for-purpose analysis followed by field testing of the product. The field performance must be monitored closely. In this interim period of field trials special permits are issued by the regulatory agencies. This approach is being used in Canada for a number of NGV cylinders involving new designs and new materials. The all composite cylinders and non auto-frettaged hoop-wrapped cylinders (tape wound under tension) are two such examples.

Even when a cylinder has been accepted within an existing code, regulatory concerns may arise due to new information which might indicate problems with material or design. Fitness for purpose in light of the new information is again used as the first step towards dealing with the problem. The project team provides these services related to the NGV cylinders in Canada.

d) Optimization of NGV Cylinders

The objective of this exercise is to introduce incremental design or material changes through collaborative effort with manufacturers. The activities outlined under a, b, and c should provide valuable insight for such incremental changes based on fitness for purpose criteria. From a marketing point of view the cylinder should be low cost and light weight and, if possible, allow for greater on board storage volumes through redesign. One or two suitable prototypes may be fabricated and laboratory tested.

CONCLUSION

This Canadian Gas Association R&D program, initiated in January 1988 and scheduled for completion in December 1989, is well underway. The technique of introducing natural cracks of known dimensions has been established. Early data on acoustic emission re-testing indicate that the hoop-wrapped cylinders are amenable to this mode of testing. Additional work on PRD elimination and redesign; and AE testing of all fibre cylinders is also under consideration.

ACKNOWLEDGEMENT

Major funding for this project has been provided by the Canadian Gas Association and Transport Canada. Co-funding in kind has come from B.C. Hydro, CNG-Fuel Systems Ltd, Faber Industries (Italy), and H.M. International (France).

Assistance from Messrs. C.T.L. Webster, D.H. Carter, J.Y. Wong, W.J. Brezden, G.E. Crombie, F. Havelock, F. McQuade and Dr. G.S. Bhuyan during the preparation of this paper is gratefully acknowledged.

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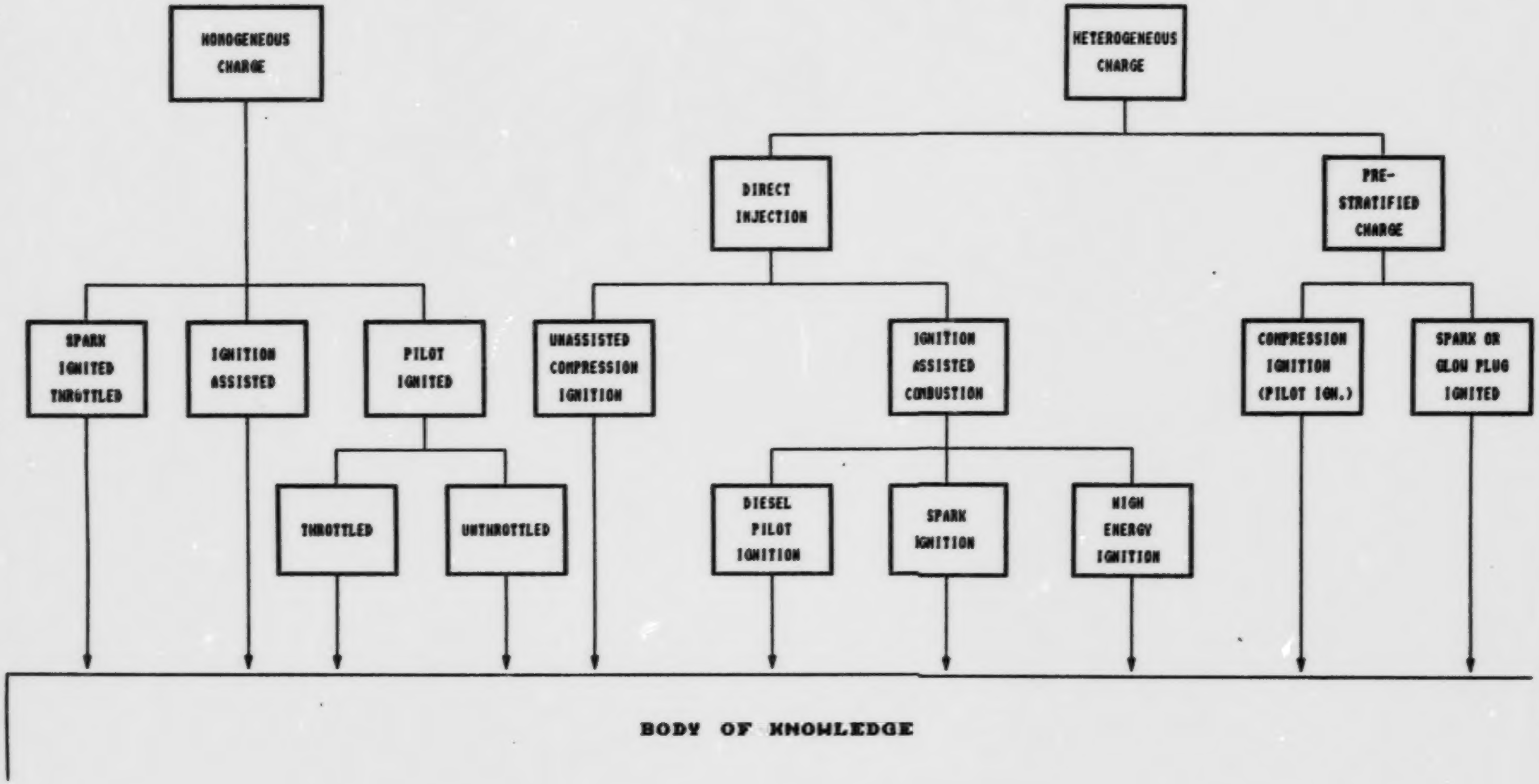
APPENDIX 5

NATURAL GAS FUELLED PRIME MOVERS

FUELLING OPTIONS


HOMOGENEOUS CHARGE
HETEROGENEOUS CHARGE
DIRECT INJECTION
PRE-STRATIFIED CHARGE
SPARK IGNITED THROTTLED
IGNITION ASSISTED
PILOT IGNITED
UNASSISTED COMPRESSION IGNITION
IGNITION ASSISTED COMBUSTION
COMPRESSION IGNITION (PILOT IGN.)
SPARK OR GLOW PLUG IGNITED
THROTTLED
UNTHROTTLED
DIESEL PILOT IGNITION
SPARK IGNITION
HIGH ENERGY IGNITION
BODY OF KNOWLEDGE

FAST BURNING IN SPARK IGNITED ENGINES
A LITERATURE SURVEY

BY

RONALD J. PIERIK
AND PHILIP G. HILL

The objective of this work is to explore the methods of increasing the burning rate of natural gas in spark ignition engines. The scope of this study is limited to spark-ignited, four stroke, piston engines.

I. INTRODUCTION
    A. Advantages of Fast Burning In Engines
    B. Disadvantages of Fast Burning in Engines
    C. Mass Entrainment Rate Equation
II. SWIRL
    A. Methods Of Swirl Generation
    B. Effects of Swirl on Fluid Dynamics
    C. Swirl Combustion In Single Cylinder Research Engines
    D. Swirling Combustion In Multicylinder Engines
    E. Bosch Swirl Chamber Spark Plug (Latsch 1984)
    F. Summary of the Effects of Swirl
III. SQUISH
    A. Thermally Optimized Porsche (Parker and Slee 1986)
    B. High Ratio Compact Chamber (Thring et al. 1982)
    C. Combined Effects Of Swirl And Squish
IV. MULTI-SPARK
V. OPENNESS
VI. TURBULENCE GENERATION
    A. Variable Valve Lift Throttling (Stivender 1968)
    B. Jet Motion
    C. Intake Passage Vortex Source
VII. COMPRESSION RATIO
VIII. SUMMARY

APPENDICES 7 - 14

7 Investigation of the Potential for Improving the Efficiency of a Spark Ignition Engine

8 Hot Rod, November 1988

9 Gas Engine Emission Controls Give New Life

10 SAE Technical Paper 872041

11 Sulzer Technical Review, 1/87

12 Wartsilavasa

13 Design of a Fast Acting Injector for Hydrogen Fueled Diesel Engines

14 APTA Conference August 14 - 17, 1988

APPENDIX 7

DOE/CE/50072-1
Distribution Category UC-96
DE86002090

Investigation of the Potential for Improving the Efficiency of a Spark-Ignition Engine Operating on Natural Gas Fuel

Final Report

October 1985

Prepared by:
Southwest Research Institute
San Antonio, Texas 78284
Under Contract No. DE-AC01-84CE50072

Prepared for:
U.S. Department of Energy
Assistant Secretary for Conservation and
Renewable Energy
Office of Transportation Systems

Alternative Fuels Utilization Program

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compression ratio while the wedge had a peak IMEP of 136 psi at the same compression ratio. This 17 percent power advantage for the hemi over the wedge arises from the hemi's better breathing characteristics. The peak IMEP for the ALDV engine was 146 psi. When the engine was converted to natural gas operation at the 8.4:1 compression ratio, the wedge combustion chamber provided only 122 psi IMEP and the hemi provided 132 psi IMEP, both well below the ALDV value. This deficiency in IMEP for natural gas operation is caused by the displacement of intake air by the gaseous fuel and the slower burn rate of the gaseous fuel. However, increasing the compression ratio to 15.5:1 restored most of the loss in IMEP for the wedge while the hemi exceeded the ALDV IMEP by a wide margin.

Knock Limits

The knock limits for wide-open throttle operation on natural gas fuel are shown in Figures 19 and 20. In these figures, three different values of spark timing are shown versus compression ratio for three operating speeds. The circle symbols denote the MBT values of spark timing; the square symbols denote the timing for which trace knock (the first evidence of abnormal combustion) occurred; and the triangle symbols denote timing for which heavy knock (large fluctuations in cylinder pressure plus noise) occurred. The onset of trace knock for the wedge combustion chamber occurred at the 14.1:1 compression ratio when the spark timing was set at the MBT value for both the 2142 rpm and 3000 rpm engine speeds. Heavy knock occurred at all speeds for the 18.5 compression ratio when the spark timing was set at the MBT value. These data show that for wide-open throttle engine operation at the 15.5:1 compression ratio, some knock (between trace and heavy) will occur when the spark timing is set at the MBT value.

The knock limits for the hemi combustion chamber, shown in Figure 20, indicate that the hemi combustion chamber was more prone to knocking than the wedge combustion chamber, again, due to the lower turbulence level in the hemi combustion chamber. Onset of trace knock occurred at the 12.7:1 compression ratio for the 2142 rpm speed with spark timing set at the MBT value. Severe knocking occurred at a slightly more advanced spark timing for the same conditions. Knock was so severe at

VII. CONCLUSIONS

Results of baseline tests showed that the shape of the combustion chamber had little effect on the equivalence ratio for best indicated thermal efficiency. For gasoline operation, the equivalence ratio for best ITE was about 0.9 and for natural gas it was about 0.8 for both cylinder heads. The slower burning rate of natural gas compared to gasoline was demonstrated by the difference in MBT spark timing. Natural gas required 10° more spark advance than did gasoline. The hemispherical combustion chamber exhibited a slower burning rate and required 10° more spark advance than did the wedge for the 8.4:1 baseline compression ratio for engine operation at the simulated 55 mph cruise condition. When the compression ratio was increased to 18.6:1, both cylinder heads required the same MBT spark timing for the cruise operating condition. The baseline peak ITE for the cruise condition was 40 percent for both heads on gasoline. It was slightly less for natural gas. The cruise peak ITE for natural gas increased linearly with compression ratio increase for both cylinder heads with the maximum ITE approaching 49 percent at the 18.5:1 compression ratio. The cruise peak ITE for natural gas was about 4.5 percentage points higher for the wedge than for the hemi for all compression ratios tested. This was caused by the effects of the higher turbulence level in the wedge combustion chamber. Exhaust emissions of hydrocarbons and carbon monoxide were about 50 percent higher for the hemispherical combustion chamber for both gasoline and natural gas. The ISHC doubled for both chambers as the compression ratio was increased to 18.6:1. However, the compression ratio had no effect on the ISCO emissions. The NO_x emissions were generally higher for the hemi and increased for both combustion chambers as the compression ratio increased. The aldehyde emissions were very low since natural gas and gasoline do not have the chemistry necessary for formation of large concentrations of these emission species. No definite trends with respect to aldehydes were found that could be related to changes in compression ratio or fuels.

The peak wide-open throttle ITE occurred at the 15.5:1 compression ratio for both cylinder heads with the wedge having a two-percentage points advantage (40 percent to 38 percent). The peak wide-open throttle IMEP also occurred at the 15.5:1 compression

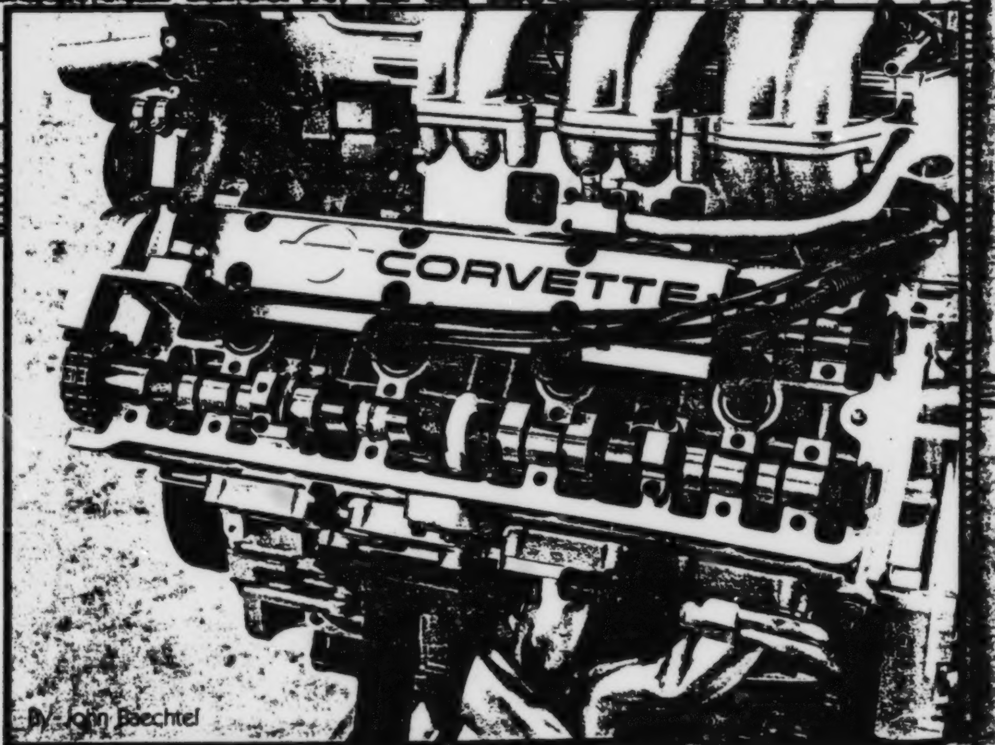
FOUR PLAY

INSIDE CORVETTE'S NEW FOUR-VALVE FOUR-CAM ALL-ALUMINUM V8

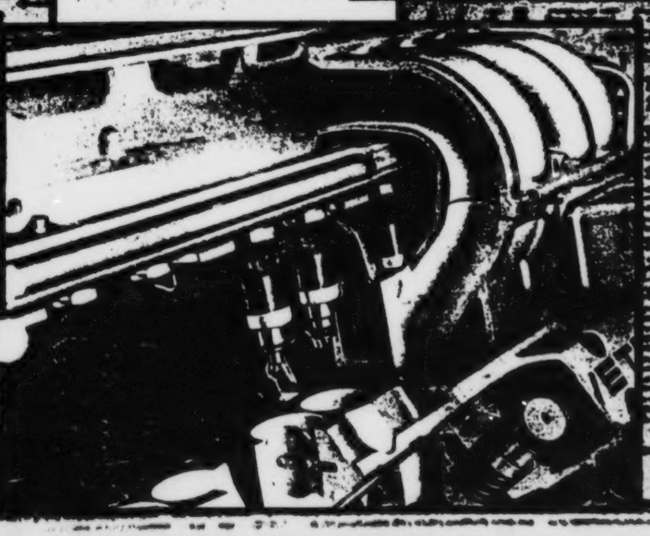
By John Baechtel

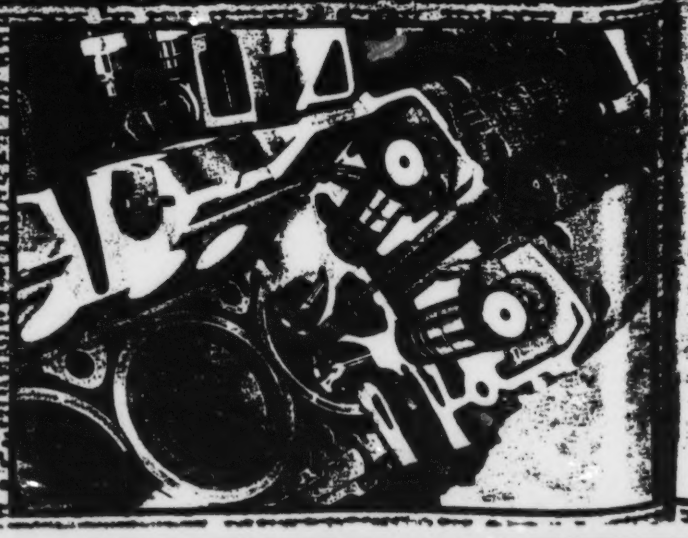
You may or may not agree with Chevrolet's decision to reinvent the small-block Chevy V8, but few people could argue that the all-new, 350-cubic-inch Corvette engine does not represent state-of-the-art, high-tech engineering. Developed jointly by CPC Engineering and GM's Lotus Division in Hethel, England, the all-aluminum, 32-valve DOHC V8 is one of the few four-cam, four-valve-per-cylinder V8 engines built anywhere for any purpose other than racing. For all intents and purposes it really is a racing engine. Refined, tamed, and emission certified, the LT5 engine slated to power the upcoming ZR1 Corvette not only shares the same general architecture as Chevrolet's highly successful V8 Indy racing engine, but it also beats the current gas-guzzler regulations and qualifies for a full General Motors extended warranty while delivering upward of 380 horsepower and 370 lbs.-ft. of torque.

The engine has already generated considerable controversy both inside and outside of GM. From the beginning it was clear that the '80s-style Corvette was destined for world-class status, and it was equally clear that the world was not ready to bestow that same status on the car's small-block Chevy powerplant, even though it has won more races and



Chevrolet's all-new 350-cubic-inch, aluminum Corvette V8 engine may be thought of as the first and only domestically produced racing engine designed for street use. With four cams, 32 valves, two injectors per cylinder, a direct-fire ignition system, and a compression ratio of 11.25:1, the LT5 delivers nearly 60-percent more power than the current 5.7L TPI small-block. This new world-class powerplant will see duty only in the special ZR1 Corvette. The four-valve design and tuned-length intake runners with electronic fuel-injection help it deliver approximately 380 hp with 370 lbs.-ft. of torque.





APPENDIX 9

USA

A breakthrough in technology for emissions control of natural gas engines used as prime movers over a wide range of industrial applications could rescue the market for natural gas as a fuel in this sphere. A contribution from Southern California.

COMPARATIVE FEATURES OF THE PSC SYSTEM WITH NSCR CATALYSTS

PSC system	Non-selective catalytic reduction
Requires no air/fuel ratio controls. Simple operation with use of a totally mechanical system and no moving parts to affect operation.	Requires expensive and complex air/fuel ratio controls to maintain system integrity and combustion efficiency.
The entire system consists of materials which are commonly available and standard throughout the industry.	Constructed of exotic metallurgy and ceramic components for durability in withstanding chemical reactions and high temperatures from the exhaust stream.
Good life expectancy of continual operation over five years and beyond in service.	Life expectancy of the catalyst element is 8,000 hours or less. System degradation takes place rapidly if air/fuel ratios are not precisely maintained.
Fuel savings have been observed ranging from 4 per cent to 8 per cent particularly ar near peak-load operation	Expected penalties in fuel consumption averaging 3 per cent.
System is versatile and not sensitive to cyclic load operation or fuel composition.	Landfill and sewage digester gases cannot be burned in engines without poisoning the catalyst. Engines which operate on varied or unsteady load conditions may fail due to rapid changes in air/fuel settings.
Regular maintenance involves only the normal replacement of engine air filters. Total service, if required, many be accomplished by general maintenance personnel using ordinary hand tools.	In addition to anticipated replacement of the catalyst element, malfunction of the controller requires service by highly skilled technicians familiar with microprocessor and electronic sub-systems.
Competitively priced with NSCR catalysts on an installed basis, with an achieved merit factor in maintenance and fuel savings.	Not overall cost effective when considering installation and downstream costs that are realised. Cost prohibitive for the smaller units.

Gas engine emission controls give new life

Natural gas engines in southern California are an important segment of the industrial market, being used as prime movers for pumping water, cogeneration, gas compression and many other industrial applications.

In recent years, however, the high cost of natural gas and the cost of emissions control has damaged the gas engine market. Electric motor manufacturers and electricity utilities have been quick to realise a potential for increasing their share, particularly in the vulnerable area of municipal and agricultural water pumping.

A breakthrough in emissions control technology could rescue the market for natural gas. It is called pre-stratified charge (psc), or selective dilution technology.

Far from being newly invented, it has been known to engine manufacturers for years, but the incentives to field test and prove its effectiveness did not develop until about three years ago when air quality management agencies imposed strict emissions standards on stationary internal combustion engines (ICEs).

Lisa Cagnolatti, environmental affairs analyst for the Southern California Gas Co, has explored the history of the natural gas engine water pumping market in a paper written to fulfil the requirements of the Chartered Industrial Gas Consultant programme — jointly sponsored by the American Gas Association and the Institute of Gas Technology.

She found that in the late 1970s, when the price of natural gas rose sharply, a special rate was established for gas used by water pump engines. This rate was beneficial to municipalities and farmers and served to maintain and even increase the market share of these engines.

Even though the price increases slowed and in many cases decreased, the cost of electicity became more competitive.

In the early 1980s electricity began making headway into the market, claiming lower maintenance costs and clean, quiet operation as major advantages. At this point, however, prices were about even with natural gas having only a marginal advantage.

On October 26, 1984 the Southern California Air Quality Management District (AQMD) adopted a new rule for emissions

continued on page 30

pipeline project is completed.

Sui Gas sells gas to Southern Gas Co which comprises two distribution utilities — Karachi Gas and Indus Gas which merged in the mid-1950s. Karachi Gas is the larger of the two, taking about three-quarters of Sui Gas's output to distribute in the Karachi metropolitan area. At present Karachi Gas supplies 400,000 domestic customers in the city area along with about 6,000 commercial customers and more than 900 industrial accounts.

Following a reorganisation of Pakistan's gas sector in the early 1960s, Indus Gas supplies customers in the southern provinces of Sind and Baluchistan, after previously supplying the whole of Pakistan outside of Karachi. The company distributes gas to about 100,000 domestic customers along with about 4,000 commercial customers and more than 200 industrial accounts.

Pakistan's recoverable gas reserves are estimated at about 16.5 tcf compared with an annual consumption of more than 320 bcf. More than 900,000 domestic customers receive piped gas throughout Pakistan, with 4,200 industrial customers and about 32,000 commercial customers.

Both the fertiliser industry and electricity sector are major consumers, each using about a third of Pakistan's production. Other industries use about a quarter and domestic customers account for about 12 per cent. At the current rate of growth, the country's reserves are expected to last more than 20 years.

The government is aware that many more customers could be signed up if more gas becomes available. In fact, some large users such as cement plants have been obliged to convert from gas to oil for all or part of their fuel requirements to prevent shortages for other smaller customers.

Most gas is produced at the Sui and Pirkor fields in Baluchistan Province. Sui is the country's largest with its estimated 5.2 tcf reserves expected to last 15 to 20 years. Its 800 mmcfd production capacity has recently been stabilised with the completion of a well head compression plant scheme by Foster Wheeler Energy Ltd.

Sui Gas is allocated 290 mmcfd of Sui's production, but the largest customer is government-run Sui Northern Gas Pipelines Ltd, which has responsibility for transmission and distribution to all areas north of the Sui field. Sui Northern is allocated 390 mmcfd and the other major customer, the Water and Power Development Authority, takes up to 180 mmcfd for power generation.

Unlike Sui Gas which sells to two utilities for distribution, Sui Northern acts as a distribution as well as a transmission company. Sui Northern's sales area includes large cities such as Rawalpindi, Multan, Faisalabad, Peshawar and Lahore, and sales are believed to have passed the 400 mmcfd mark to 410,000 residential, 22,000 commercial and 3,100 industrial customers.

Sui Northern buys most of its gas from the Sui field, also taking a small amount of associated gas from the Meyal and Toot fields in northern Pakistan. The company recently used World Bank funding to expand its transmission and distribution grid to boost its system capacity from 390 mmcfd to about 490 mmcfd.

The Pirkoh gas field 70 km north of the Sui field is Pakistan's other major source. Recoverable reserves are 2 tcf, and estimated reserves between 3.8 tcf and 11 tcf.

Pirkoh production had recently been increased from 100 mmcfd to 180 mmcfd with Asian Development Bank assistance, and most of the additional gas has been allocated to Sui Gas which is allowed to take up to 110 mmcfd from the field.

The Mari field is also an important producer. Located on the Indus River right bank 80 km south-east of the Sui field in Sind Province, its recoverable reserves of 3.6 tcf are expected to last another 25 years.

Because of its composition, Mari gas is entirely used by three fertilizer plants within a radius of 50 km. World Bank funding has been used to boost the field's production level from 200 mmcfd to about 380 mmcfd.

Pakistan wants to develop smaller associated and non-associated gas reserves. Falling reservoir pressure caused production to stop at the Sari and Hundi gas fields near Karachi in the early 1980s, but Pakistan's Oil and Gas Development Corp intends to reactivate them.

The corporation has requested a US$ 35 million loan from the Asian Development Bank to redevelop Sari and Hundi, the nearby Khotar field, and Nandpur about 180 km north-east of Sui. It is hoped that Sari and Hundi will deliver a combined output of 15 mmcfd.

The Oil and Gas Development Corp also has asked the Asian Development Bank for a technical assistance loan to hire consultants to prepare a project loan application to develop the Uch field about 50 km south of the Pirkoh field. The corporation wants a gas reserve and gas utilisation study for the low calorific value reserves and one early suggestion is that Uch be set aside for power generation. ●

Gas engine emission controls give new life

continued from page 28

from stationary internal combustion engines. These strict NOx reduction regulations made necessary expensive retrofits for many engines because selective catalytic reduction was the only proven technology for reducing NOx.

Faced with the prospect of losing a large portion of its gas engine market, the Southern California Gas initiated a testing project for pre-stratified charge.

Psc systems incorporate a technique called selective dilution to control combustion processes in rich burn engines, limiting the formation of NOx in the cylinder. By limiting the formation of the target pollutant, no after-treatments or catalysts are required.

The installation of a retrofit psc system does not involve expensive internal modifications to engines. The only modification is to inlet manifolds and passive adjustments to air/fuel ratio and spark advance for economical engine operation. The process is self-activated by the basic principle of the four-stroke engine.

The system comprises a set of tube-like devices called 'straws' in each port of the intake manifold. One of these straws directs filtered combustion air into the port to fill the region near the intake valve of each cylinder. The second straw directs a small amount of relatively inert recycled exhaust gas into a layer between the air and the fresh charge which fills the rest of the manifold. These layers are arranged before the gases are introduced into the cylinder, hence the name 'pre-stratified charge'.

The step-by-step cycle is:

- With the intake valve to a cylinder closed, the air flow in the port and manifold runner comes to rest for about three-quarters of the four-stroke cycle while the remaining cylinders are filing with fresh charge.
- During that time, the next air charge to this cylinder is pre-stratified, or 'pre-arranged' in the port by the dilution straws. This arrangement yields a mixture of fresh charge which is separated from the dilution air by a thin layer of inert gas (recycled exhaust gases). As a result, the engine retains the simplicity of rich-burn while obtaining the relevant advantages of overall lean-burn operation.
- When the intake valve opens, the pre-stratified mixture enters the cylinder first, followed by fresh charge from the carburettor or mixing valve.
- The compression stroke occurs rapidly, so there is very little time for the charge to mix and destroy the stratified layers. An easily ignitable fresh charge mixture remains near the spark plug.
- Upon combustion, the high dilutions 'cool' the burning gases. Also, the thin layer of inert exhaust gas displaces the excess oxygen (needed to form NOx) away from the combustion zone, and less NOx forms with the psc system than otherwise. The cooling effect of the psc system substantially reduces the risk of detonation, so the operator can advance the spark timing for improvements in fuel economy. ●

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High-Pressure Late Cycle Direct Injection of Natural Gas in a Rail Medium Speed Diesel Engine

James F. Wakenell
Southwest Research Institute
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Quentin A. Baker
Wilfred Baker Engineering

International Fuels and Lubricants
Meeting and Exposition
Toronto, Ontario
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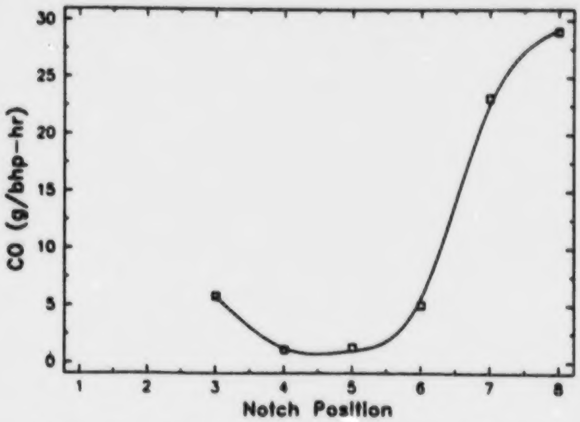

Figure 15. Brake Specific Carbon Monoxide Emissions

increase from under 5 to over 20 g/bhp-hr. Brake specific hydrocarbon (HC) emission level was very high at Notch 3, as shown in Figure 16. This indicated that the unstable operation observed at this lower notch position was a result of poor combustion from a very lean fuel/air ratio.

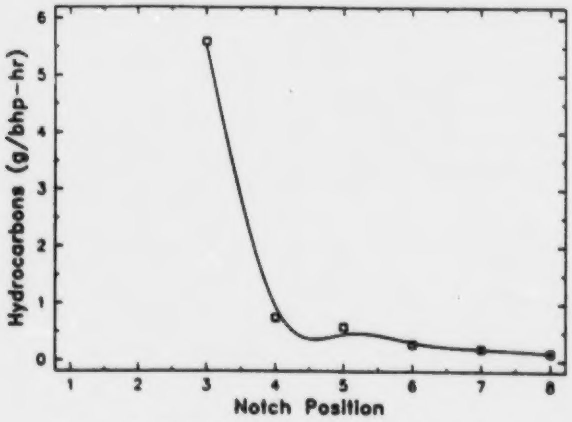

Figure 16. Brake Specific Hydrocarbon Emissions

CONCLUSION

In general this project achieved its objective, and that was to prove the feasibility of high-pressure/late-cycle natural gas injection in a medium-speed locomotive engine. Rated speed and power were achieved without reducing compression ratio. This is impossible with low-pressure injection of natural gas. It was also demonstrated that operation at the lower notches is possible where the low-pressure method reaches a lean flammability limit. With optimization of the injectors and injection system, there is no doubt that full notch operation, including idle, would be possible, with performance equal to or slightly below baseline diesel fuel levels.

RECOMMENDATIONS

Although the feasibility of natural gas operation in the medium-speed diesel engine was proven, much work remains to be done with optimization of the injection system. Better placement of the pilot injectors to take full benefit of the gas spray would ensure reliable and efficient ignition of the gas. Gas and pilot injectors with sufficient turn-down ratios would enable stable operation at the lower notches and idle. Optimization of the combustion process would undoubtedly shorten the injection process, resulting in higher thermal efficiencies. Finally, work could be done in the area of spark or glow plug ignition of the natural gas, eliminating the need for dual fuel.

ACKNOWLEDGEMENTS

The authors wish to acknowledge that this project was funded by the Department of Energy through a subcontract with Martin-Marietta Energy Systems (Oak Ridge National Laboratory). The DOE program manager was Gene Ecklund, and the Martin-Marietta technical contract manager was Ron Graves. We wish to thank these individuals for their support and cooperation and for permission to publish these findings.

The authors also wish to acknowledge the efforts of Wesley Robert Shultz in running the engine and support equipment during LNG tests.

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APPENDIX 11

RTA Dual-Fuel Engine – Natural Gas Instead of Diesel Oil

B. Engesser, Diesel Engine and Diesel Power Plant Division

2175

Testbed trials of the Sulzer RTA84 dual-fuel engine were successfully completed at the IHI Aioi engine works (*Japan*) in April 1986. This newly-developed engine-type, output range 15 to 40 MW, can be operated with both diesel or heavy oil and methane gas at comparable thermal efficiency and unchanged output. The RTA dual-fuel engine was developed in close collaboration between Sulzer Brothers Limited, Switzerland, and its Japanese, licensees Ishikawajima-Harima Heavy Industries Co. Ltd. (IHI), Nippon Kokan KK (NKK) and Sumitomo Heavy Industries Ltd. (SHI). Intended for the propulsion of LNG carriers, where the boil-off gas from the ship's cargo is exploited or for the generation of electricity in stationary plants, the RTA dual-fuel engine is an economic and reliable alternative to the steam or gas turbine.

1 The LNG tanker "Venator", with her 7RNMD90 engine, has provided valuable experience of a large dual-fuel engine and its operation in a shipboard environment.

Dual-Fuel Engines – not new as far as Sulzer is concerned

SULZER developed a dual-fuel engine, Type RND90M, already in the 1960s. This functioned using the so-called low-pressure gas injection principle. The fuel gas is admitted into the cylinder during the scavenging phase, compressed together with the air charge and fired by injecting a small amount of diesel or heavy oil. Knock, the main problem associated with this engine, could only be prevented by measures causing a lowering of the mean pressure and efficiency compared with the version running on diesel oil alone.

The Venator, a 29000 m^3 LNG carrier (*Fig.* 1), was equipped with a 7RND90M dual-fuel engine in 1972 and is still in service.

New concept raises the economic performance

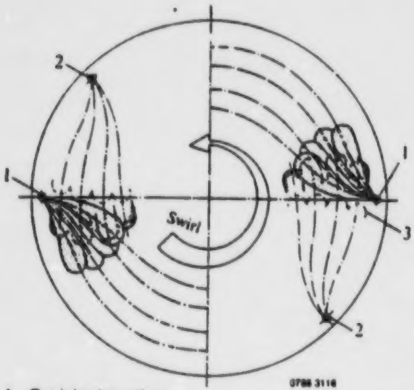
Brought about by the oil price increases of the 1970s, diesel engine development over the last decade has been focussed mainly on bringing down the fuel consumption. A parallel trend to higher power-to-weight ratios pushed up the effective mean pressures considerably.

These new constraints, valid also for a dual-fuel engine in keeping with the times, led to the search for a new concept, which finally emerged as the high-pressure gas injection principle.

In this case, the fuel gas is compressed to about 250 bar and blown into the cylinder at the end of the compression stroke. In this way, the danger of knocking is eliminated, and the way is opened for a modern engine design. With this system also, combustion is initiated by injecting a small amount of diesel or heavy fuel oil.

Design specification

Not only the operating data and operating range of the RTA dual-fuel engine were defined in the specifica-



2
1
Swirl
1
3
2
0788 3116

1 Gas injection valves
2 Pilot injection valves
3 Pilot jet front at time of maximum combustion intensity

2 Combustion concept for the Sulzer RTA dual-fuel engine. By using separate oil injection nozzles and exploiting the combustion air swirl, optimal ignition is made possible even with small amounts of pilot oil.

tions, but also the basic guidelines for the design. This was to ensure that the safety aspects, which claim priority where methane gas is being handled, were taken into account already at the concept stage.

Operating data and operating ranges:

- Mean pressure and fuel consumption comparable to those of standard diesel engine
- Dual-fuel operation possible over load range 25 to 100%
- Pure diesel or heavy fuel oil operation possible up to 75% load
- Operation possible at full load with pilot oil less than 10% of total fuel input.

Guidelines for design:

- As few modifications as possible to the standard RTA engine
- Retention of as many proven components as possible
- Clear-cut separation of functions for all modified or new components
- Priority for security against gas leakages.

Principal features of the RTA dual-fuel engine

Combustion concept (*Fig. 2*)

Located in the cylinder cover at the periphery of the combustion chamber are two pilot oil and two gas injection valves. Assisted by the swirl in the combustion air induced during scavenging, the pilot oil jets injected first, already burning at the front, meet the injected gas jets a few milliseconds later, so that these in turn begin to burn. The angle between pilot oil and gas nozzles, measured in the direction of the swirl, is chosen so that the flame of the pilot jet reaches its highest intensity at the moment of impinging upon the gas jet.

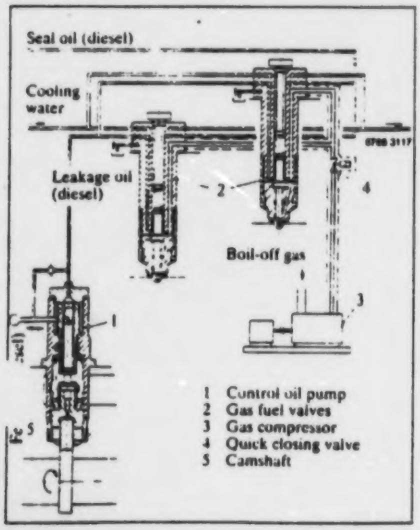
Pilot oil injection system

It was possible to retain the oil injection system of the standard engine in its essentials. Only the outlet cross sections of the nozzles and the plunger diameter had to be adapted to the new conditions, so as to assure proper atomization of the pilot oil at all operating points.

Gas injection system (*Fig. 3*)

The fuel gas is led to the gas valves through a double-walled piping system under a constant pressure of 250 bar. A nozzle needle, pressed onto its seat in the rest position, can be lifted by a hydraulic pressure pulse, allowing the gas to be blown in. This pressure pulse is delivered by a helix-type injection pump derived from the ZA40 four-stroke engine. Pulse onset and duration can be varied by means of a control linkage. This arrangement thus enables start of the injection and fuel gas quantity to be controlled.

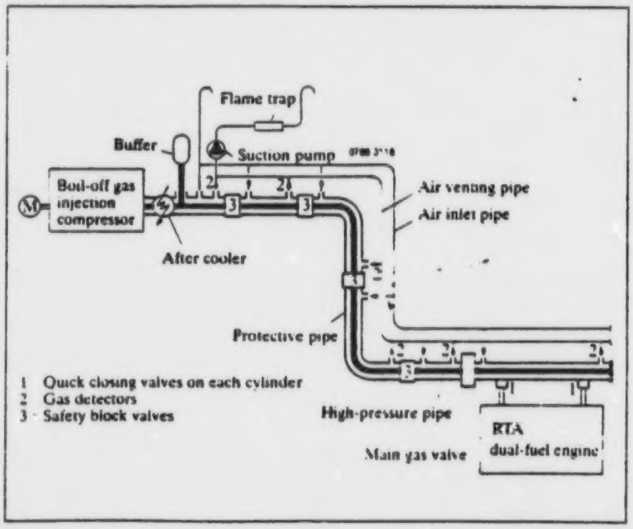
A sealing oil system prevents gas leakages at all points imaginable on the gas valve. The parts of the gas valves projecting into the combustion chamber are watercooled, in order to prevent overheating in pure diesel service.



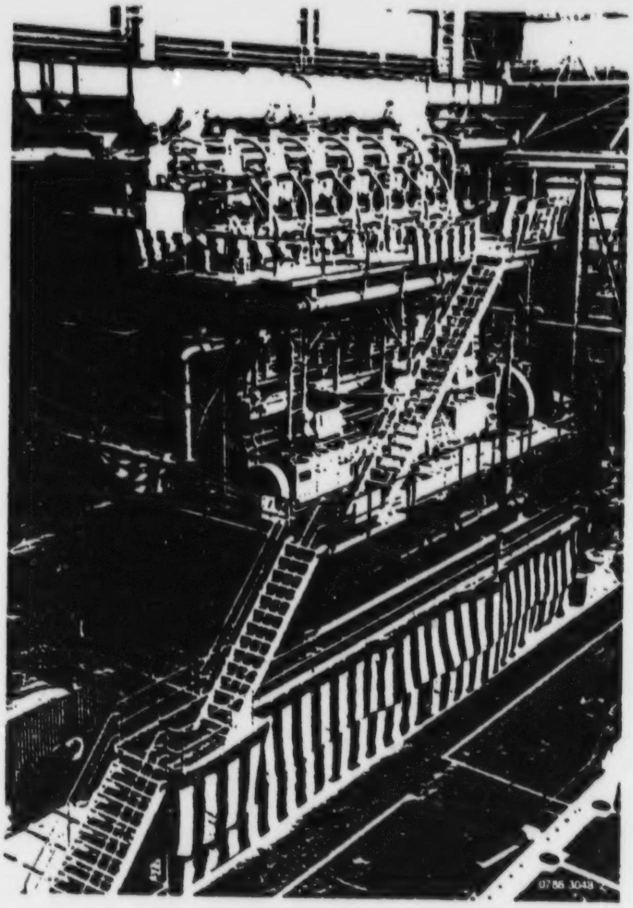
Seal oil (diesel)
Cooling water
0788 3117
Leakage oil (diesel)
2
4
Boil-off gas
3
1
5
1 Control oil pump
2 Gas fuel valves
3 Gas compressor
4 Quick closing valve
5 Camshaft

3 Gas injection system for the RTA dual-fuel engine. A hydraulic pressure pulse is generated by the control pump, effecting controlled opening of the nozzle needle.

4 Layout of the safety elements for the high-pressure gas supply system.



Flame trap
Buffer
Suction pump
0788 3118
Boil-off gas injection compressor
M
2
2
3
3
Air venting pipe
Air inlet pipe
After cooler
Protective pipe
2
2
2
3
1
1
1 Quick closing valves on each cylinder
2 Gas detectors
3 Safety block valves
High-pressure pipe
Main gas valve
RTA dual-fuel engine



5 6RTA84 experimental engine at IHI with cylinder modified for dual-fuel operation. This was a standard production engine with 2100 kW cylinder output at 66 rev/min.

Control system

The governor of the standard engine retains its function. An electronic control unit apportions the pilot oil and gas feeds. It is connected via control linkage with the pilot oil injection pump and fuel gas control pump, in such a way that no uncontrolled operating state can result from a failure or malfunction of the electronics.

Both injection systems are provided with separate emergency shutoff devices. Via the electronic control unit, the pilot oil and gas proportions may be adjusted manually or automatically according to the available gas supply.

Safety system

The safety system covers not only all gas-conveying elements on the engine, but also includes the entire feed line from the high-pressure gas compressor (*Fig. 4*). The layout may be summarized as follows:

- All gas-conveying components embody a high degree of security against uncontrolled gas leakages inherent to the concept
- Gas leakages can be detected and the gas feed can be shut off long before the gas concentration reaches explosive levels at any place
- By dividing all high-pressure gas pipes into sections, which are isolated at once by self-actuated shutoff valves in the event of pipe rupture, any gas escape is held within very narrow limits
- All high-pressure gas pipes are surrounded by a protective pipe. Air flows constantly through the cavity formed by this, allowing speedy leak detection by means of gas detectors. The cavity volume has been selected so that, in the event of a break in the high-pressure gas pipe, the escaping gas can be accommodated without excessive pressure rise.
- Quick shutoff valves in the high-pressure gas supply pipe in the immediate proximity of the gas injection valves can be actuated if the injection rate is too high.
- An electronic security system monitors the gas concentrations at the venting points of the protective cavities, the gas pressure, sealing oil pressure, motion of the gas valve nozzle needle and the amount of energy liberated in the combustion chamber. If a fault is detected, the appropriate safety device is activated.

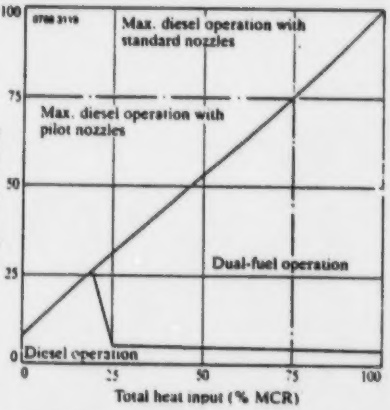
Trials – step by step to the prototype

The SULZER dual-fuel concept was demonstrated on one cylinder of a 6RTA84 engine in April 1986 (*Fig. 5*).

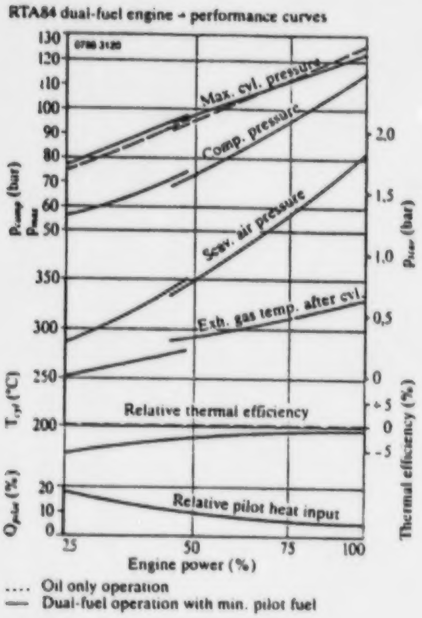
This demonstration was preceded by a series of component tests, checking of functions and interaction of the systems on the prototype engine with simulated dual-fuel operation, and finally the actual tests on this engine.

Component tests:

- The entire gas injection system was rigged on a test stand. Intensive tests performed using compressed air instead of methane gas.
- Function tests of the electronic systems together with sensors and actuators. Simulation of all operating states possible.



6 Correlation between shaft power and heat input for the RTA dual-fuel engine. Between 25% and 100% load, dual-fuel operation is possible with a minimal pilot oil input amounting to 5% of the total fuel input at full load.



7 Characteristic data of the RTA84 dual-fuel engine. Gas operation with minimal pilot oil injection yields values roughly equivalent to pure diesel operation.

Engine trials:

Reference measurements in pure diesel operation with same injection equipment for all cylinders.

- Flame front propagation measurements on the ignition jet to determine the optimal timing for gas injection.
- Simulated gas injection using compressed nitrogen (250 bar) to confirm the dynamic behaviour of the gas injection system on the engine.
- Function tests of the safety system.
- First dual-fuel operation to confirm the combustion concept and for fine adjustment of the oil and gas injection timings, control linkage and electronic control unit.
- Determination of the minimum pilot oil feed quantity at which stable hydraulic behaviour of the oil injection system is still assured.
- Variation of the pilot oil and gas proportions, together with measurement of their influences on combustion.
- Variation of the gas jet directions and measurement of the effects on the surface temperatures of the combustion chamber components.

Variation of the feed pressure and nitrogen content of the fuel gas to determine their influences on the combustion process.

Test results

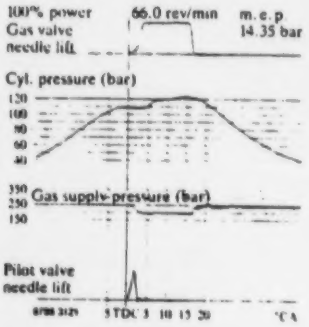
The trials showed that the system selected was fully adequate to the requirements laid down in the specifications (*Figs. 6* and *10*). As far as the following points were concerned, these were even exceeded:

- Irreproachable combustion was demonstrated at full load with the pilot oil feed reduced to 5% of the total fuel input.
- By further reduction of the pilot feed to nil, quality of combustion was impaired more and more, but a failure of combustion could never be recognized.

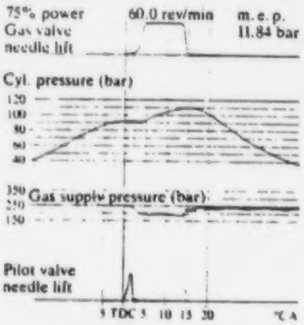
Throughout the testing phase, the running behaviour of the engine was unaccompanied by problems or special occurrences. All newly developed components displayed perfect appearance at the end of the trials.

Summing-up

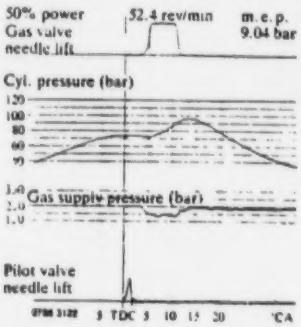
The engine trials demonstrated that operating data comparable to those of a standard engine may be expected using high-pressure gas injection on a modern, two-stroke, large-bore diesel engine. Consequently, the RTA dual-fuel engine offers a reliable alternative yielding high output for low fuel consumption where natural gas is available. Ω



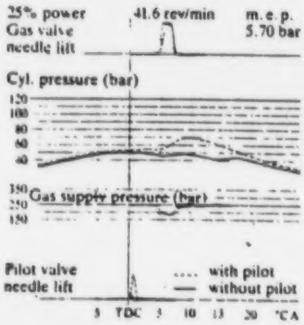
8 Injection timings and cylinder pressure for dual-fuel operation at 100% load and minimal pilot oil rate.



9 Injection timings and cylinder pressure for dual-fuel operation at 75% load and minimal pilot oil rate.



10 Injection timings and cylinder pressure for dual-fuel operation at 50% load and minimal pilot oil rate.



11 Demonstration of self-ignition of the inblown gas at low load when the pilot oil injection is virtually interrupted. Combustion proceeds much slower, and the maximum cylinder pressure is much lower than with pilot oil.

APPENDIX 12

WARTSILA VASA
GAS DIESEL
ENGINES

WÄRTSILÄ DIESEL

WÄRTSILÄ VASA GAS DIESEL ENGINES

PERFORMANCE

The high pressure gas injection method obtains full benefit from a modern diesel engine. The same high output and efficiency is available, with either gas or diesel oil. The heat balance and thermal and mechanical load on the engine do not differ significantly when switching from a gaseous to a liquid fuel.

SPECIFIC FUEL CONSUMPTION

BSFC apply for ISO 3046/1 reference condition and without engine driven pumps. Tolerance + 5%.

VASA 22 GD, 900 rpm/1000 rpm*, VASA 22/26 GD, 750 rpm

Load (%)	Diesel mode (g/kWh)	Gas mode Oil (kJ/kWh) + Gas (kJ/kWh)	Total (kJ/kWh)
100	189	400 + 7670	8070
85	189	480 + 7590	8070

VASA 32 GD, 720 rpm/750 rpm

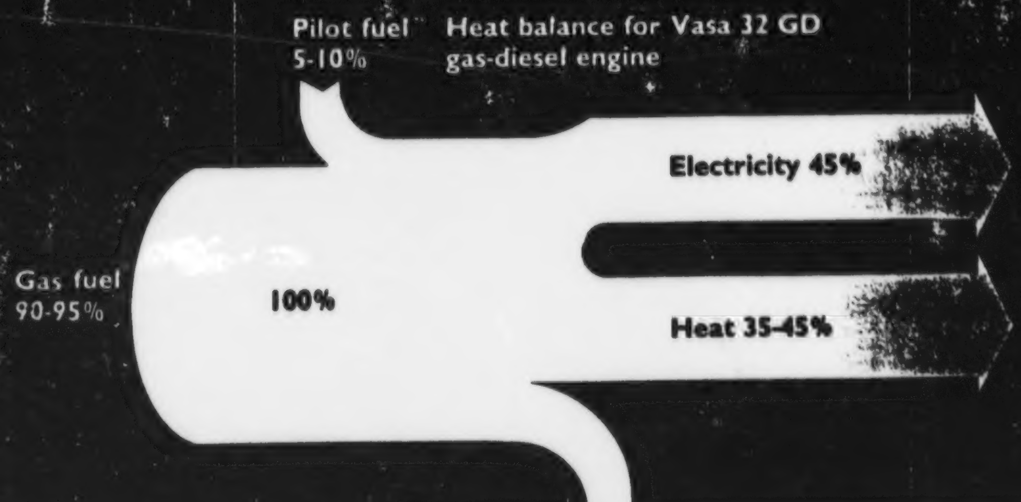
Load (%)	Diesel mode (g/kWh)	Gas mode Oil (kJ/kWh) + Gas (kJ/kWh)	Total (kJ/kWh)
100	181	400 + 7330	7730
85	181	480 + 7250	7730

VASA 46 GD, 450 rpm/500 rpm/514 rpm

Load (%)	Diesel mode (g/kWh)	Gas mode Oil (kJ/kWh) + Gas (kJ/kWh)	Total (kJ/kWh)
100	173	400 + 6990	7390
85	170	480 + 6780	7260

[*] BSFC 3 g/kWh higher

Heat balance for Vasa 32 GD gas-diesel engine



COMPUTER BASED CONTROL/SAFETY SYSTEM

An electronic governor is used for maximum flexibility. The optimal operating mode, diesel or gas, can be chosen by pushing a button.

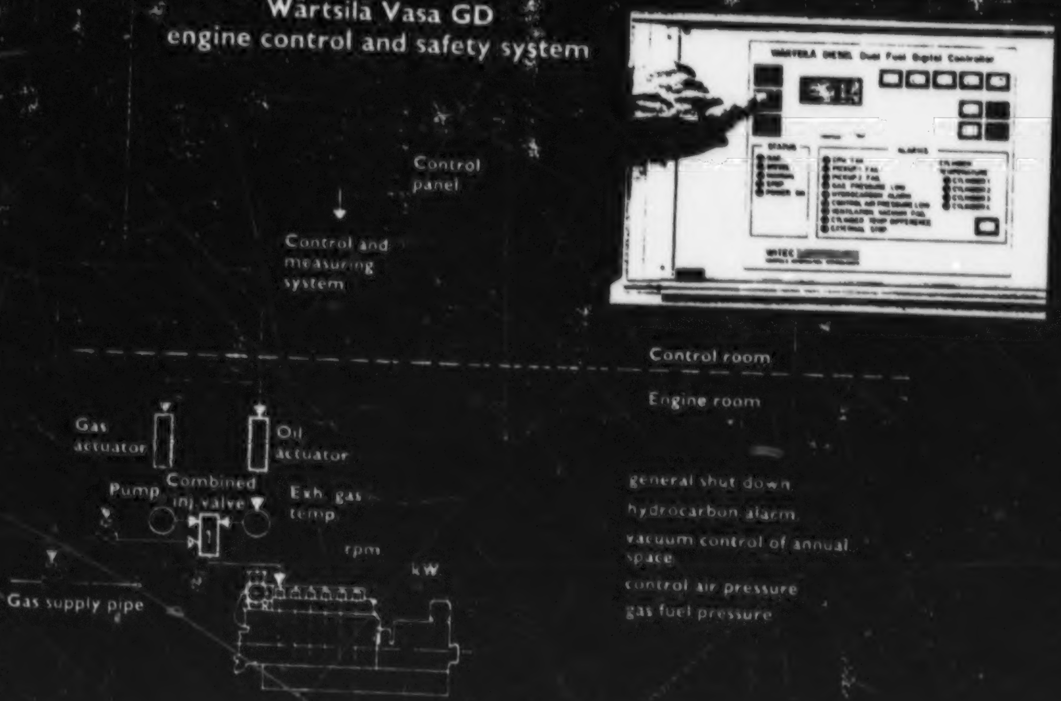
■ "GAS":	a preprogrammed gas diesel ratio will be maintained.
■ "DIESEL":	normal diesel oil or heavy fuel operation.
■ "MANUAL":	the desired gas diesel ratio can be adjusted from the control panel.

The changeover from gas-to-diesel mode is automatic if the gas supply is disrupted.

Start and low load operation are always carried out on diesel oil, even with the control set in GAS mode.

The safety system is also an essential part of the design concept. Gas engines are fitted with both the normal safety system mounted on conventional diesel engines and additional safety devices. The safety system must also meet the requirements of classification societies for marine installations and local regulations for land-based power plants. The main features of the concept are as follows:

- ■ A double walled pipe system eliminates gas leakage to the engine room.
- ■ An automatic main gas valve in front of each engine and a quick closing valve mounted at each cylinder shut off the gas supply in the event of:
  - • excessive gas flow to any cylinder
  - • disturbance in the load balance of the engine
  - • gas leakage to the annular space
  - • low output from the engine
  - • low gas supply pressure



Wärtsila Vasa GD engine control and safety system

WÄRTSILÄ DIESEL Dual Fuel Digital Controller
Control panel
Control and measuring system
Control room
Engine room
Gas actuator
Oil actuator
Pump
Combined inj. valve
Exh. gas temp.
rpm
kW
Gas supply pipe
general shut down
hydrocarbon alarm
vacuum control of annual space
control air pressure
gas fuel pressure

WARTSILA VASA
GAS
DIESEL
ENGINES

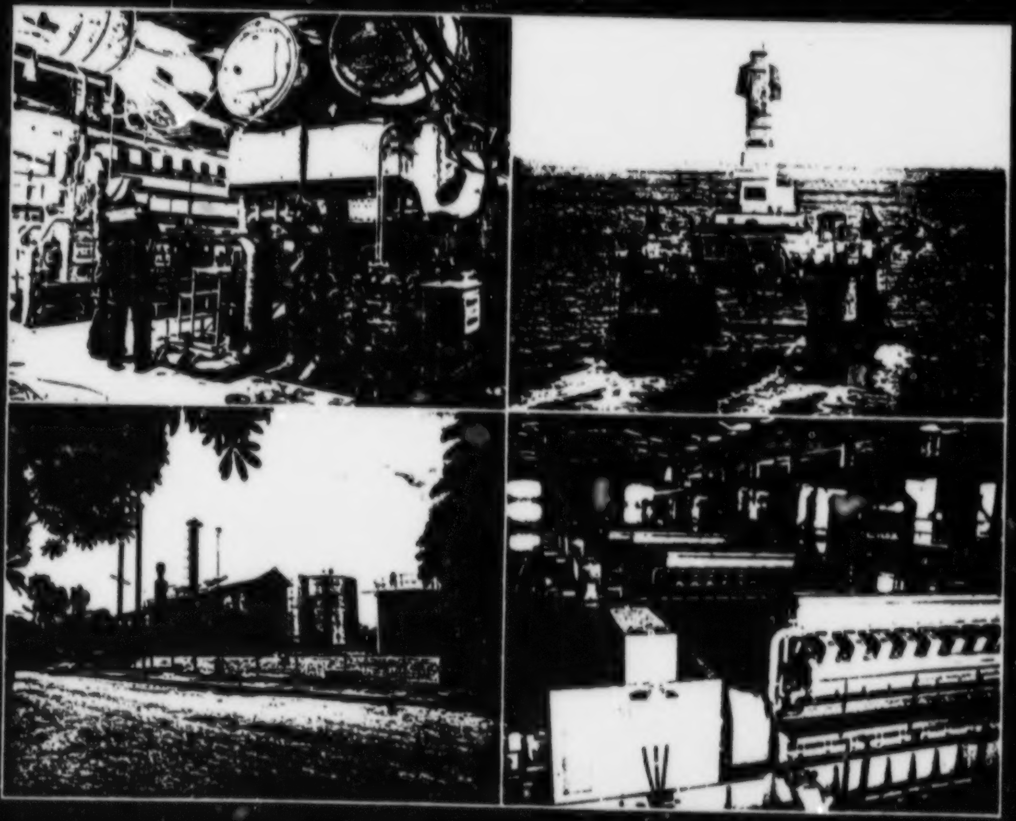
FUEL INJECTION SYSTEM

The compressed gaseous fuel is transported from the gas compressor, which is normally located outside the engine room, to the injection valve on the engine through a double-walled pipe system. The annular space is mechanically vented to a safe place outside the engine room, where the air is checked constantly by hydrocarbon detectors. The key element in the design concept is the combined injection valve, through which both the pilot fuel and the gaseous fuel is injected. The same valve can be used for operation on gaseous fuel with pilot injection or for diesel fuel alone up to full output in both modes. A special injection valve design makes it possible to run the engine continuously on heavy fuel without causing any risk of damage to the gas admission side. Gas mode operation requires a minimum 5% of MCR consumption as pilot fuel. This amount will be constant within the range above 30% load.

APPLICATIONS

Wartsila Vasa engines are widely used in various marine and landbased applications. The same market area can be targeted with the gas-diesel engine concept provided that gas fuel is available. The largest potential will be found in power plant applications where cogeneration of power and heat is feasible and on different types of floating production/drilling vessels. Offshore production facilities may become completely self-sufficient with respect to fuel if the crude oil can be used as pilot oil and back up fuel. Other areas of interest are as a primemover for pumps, compressors and mills.

A great advantage of the new gas-diesel engine concept is that existing installations using Vasa type engines for diesel or heavy fuel oil operation can be equipped with the gas injection system, without any modifications to the basic engine.



DESIGN OF A FAST-ACTING INJECTOR FOR HYDROGEN-FUELED DIESEL ENGINES

Phase II

Final Report

to

Transportation Energy Branch
Energy, Mines and Resources Canada

by

Chris J. Green
and
James S. Wallace

Centre for Hydrogen and Electrochemical Studies
University of Toronto

August, 1988

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1. PROBLEM STATEMENT

A recent study of a hydrogen-fueled diesel engine demonstrated increased power output compared to the same engine operated on diesel fuel [1]. While this is a promising result, the part load efficiency was not as high as expected. The injection duration was believed to be greater than that required for optimum efficiency. However, the injector used, a hydraulically actuated unit, did not have the range of injection durations needed to achieve optimum efficiency.

The present project was undertaken to design a fast acting hydrogen ~~emission~~ admission valve (injector) which would allow a more thorough investigation to optimize the part load efficiency of the hydrogen fueled diesel engine. The major design target is that the duration of injection be dynamically controllable over the range 0.5ms to 6.0ms.

2. DESIGN CRITERIA

The following requirements established the design boundaries:

- injection duration dynamically controllable from 0.5ms to 6.0ms.
- dynamically adjustable injection timing.
- able to supply hydrogen to the cylinder at 10.3 MPa (1500 psi).
- able to remain closed when subjected to cylinder pressures of 15 MPa (2200 psi).
- able to withstand internal pressures of 20 MPa (2900 psi).
- produce an instantaneous flow rate of 6176 SLPM of hydrogen when open (average of 343 SLPM).
- operate at temperatures between 10 C and 100 C.
- be fail safe in operation. i.e. failure in actuator power should result in a closed valve.
- capable of operation from an automotive electrical system.
- be producible with standard machining techniques.

- be amenable to performance monitoring such as needle lift times.
- allow for easy modification of injector tip geometry.
- have a suitable life or mean time between failures for engine testing.
- actual sizing to allow for installation in a single cylinder Lister diesel engine.

3. DESIGN ALTERNATIVES

Three approaches to achieving the design goal were examined: piezoelectric actuation, electrohydraulic actuation and electromagnetic actuation.

A piezoelectric material is one which undergoes a dimensional change when subjected to an electric field. Such materials produce high force output and fast response times, however, the actual displacements are quite small. A mechanical linkage or hydraulic amplifier can be used to increase the displacement but this will reduce the applied force. A relatively high voltage, on the order of 500 - 1000 volts is required to drive the piezoelectric actuator. This requirement coupled with the initial high current demand (due to it being a capacitive load) pose difficult design problems since the driver electronics should be able to operate from the 12 VDC vehicle electrical system. A further difficulty with the piezoelectric actuator is that the material has a thermal expansion coefficient close to $4\times10\text{-}6\ C^{\circ -1}$. Consequently even relatively small temperature changes produce an effect similar in magnitude to the electrical stroke [2]. These problems along with the requirement for very high precision machining needed to take advantage of the small piezoceramic displacements made this approach unsuitable.

The electrohydraulic option for actuation of the pintle was to use an hydraulic piston integral with the pintle and preloaded on both sides. Actuation would be accomplished by release of the hydraulic preload from one side of the piston by a solenoid relief valve. Relief of the preload from the opposite side of the piston would allow a return spring to close the pintle. This strategy used the hydraulic circuit to magnify the actuating force produced by the solenoids. The disadvantages associated with this force multiplying strategy are:

APPENDIX 14

USE OF IGNITION IMPROVING ADDITIVES IN METHANOL TRANSIT CONVERSIONS

-- PERFORMANCE AND COST IMPLICATIONS --

for

THE AMERICAN PUBLIC TRANSIT ASSOCIATION
ALTERNATIVE FUELS/EXHAUST EMISSIONS WORKSHOP

TACOMA, WASHINGTON

August 14 - 17, 1988

THREE 1964 6V-71 POWERED, 35-FOOT GMC BUSES HAVE BEEN OPERATED IN JACKSONVILLE, FLORIDA FOR THE PAST SIX MONTHS ON METHANOL, ENHANCED WITH AN IGNITION IMPROVER -- *AVOCET*

- PROBLEMS WITH CASTOR OIL PROMPTED INITIAL INVESTIGATION OF ALTERNATIVE ADDITIVES
    - INJECTOR TIP CLOGGING
    - ACCELERATED ENGINE WEAR DUE TO STICKING RINGS/COATING OF COMBUSTION CHAMBER
- SEVERAL ALTERNATIVE LUBRICANT PACKAGES WERE INVESTIGATED; AVOCET WAS CHOSEN FOR ITS IGNITION IMPROVING PROPERTIES AS WELL AS FOR LUBRICANT AND CORROSION RESISTANCE PROPERTIES
- PERSISTENT AND SEVERE PROBLEMS WITH GLOW PLUG RELIABIITY AND DURABILITY MADE THE AVOCET OPTION ESPECIALLY ATTRACTIVE

APPENDICES 15 - 21

15 High Energy Ignition Systems

16 Review, January 1988

17 SAE Technical Report 880149

18 Gas Engine Durability Improvement

19 SAE Technical Report 880494

20 Advanced Natural Gas Fueled Engine

21 FEV of America, November 1988

APPENDIX 15

UNIVERSITY OF VICTORIA - PLASMA PHYSICS GROUP

SUBCONTRACT - PROJECT 1.1.5 - HIGH ENERGY IGNITION SYSTEMS

FINAL REPORT FEBRUARY 1986

SUMMARY OF YEAR'S RESEARCH AND DEVELOPMENT

A survey of existing high energy ignition concepts that have been proposed for use in internal combustion engines was undertaken. Candidate systems included: plasma jet ignition, puff-jet ignition, ultra-fast spark ignition, and multiple electrode spark ignition. Plasma jet ignition was chosen as the most promising system to incorporate in a multi-cylinder natural gas fuelled vehicle on the basis of its proven potential for reduction of ignition delay and the enhancement of initial burning rates of methane-air fuel mixtures.

Initially a single ignition source system was designed, incorporating a General Motor's ignition module, with the capability of expansion into a six cylinder version. This was first tested in a constant volume combustion bomb and then in a single cylinder L-head engine. Results of this new system confirmed previous work on the engine showing a significant reduction in ignition delay, an extension of the lean mis-fire limit and a slight reduction in combustion time. The engine's output power was substantially increased for lean mixtures.

A six cylinder version of this ignition system was then developed and the plasma plugs break down gap shortened to achieve reliable break down at the higher compression ratios experienced in the Chevrolet S10 test vehicle. While the vehicle would run satisfactorily under no load conditions when fuelled by either gasoline or natural gas, and under load when fuelled by gasoline, the load performance of the engine when fuelled by natural gas was erratic.

A mathematical model developed during the last year analyses this problem in terms of the fluid dynamics of the plasma jet plume and suggests modifications which may resolve it.

APPENDIX 16

CENTRE FOR ALTERNATIVE TRANSPORTATION FUELS REVIEW

January 1988

Issue No 7

IN THIS ISSUE

- *Diesel Emissions*
- *CGA Research Update*
- *Chev 454 N.G. Engine*
- *Propane Cummins 855*

RESEARCHERS SEEK SOLUTIONS TO HEAVY DUTY ENGINE EMISSIONS

Manufacturers of heavy duty diesel engines for trucks and buses are currently concentrating their research on how to meet the new emissions regulations to take effect in the U.S. and Canada in 1988, 1991, and 1994. At **B.C. Research** and at the **Ontario Research Foundation**, approaches using natural gas are being tried.

The new regulations limit the emissions of Carbon Monoxide (CO), unburnt hydrocarbons (HC), oxides of nitrogen (NOx), and particulates from engines used in heavy commercial road vehicles. Typically, these engines are diesels ranging from 6 to 12 litres displacement, and developing from 200 to 600 horsepower.

The first three pollutants, CO, HC, and NOx, have long been limited by legislation applying to automobile spark ignition (SI) and diesel (CI) engines. Particulates, identified as a component of diesel smoke, smog, and a carrier of carcinogens to the lungs, are now attracting much stricter regulation than previously.

Production of Pollutants

The ideal combustion of a hydrocarbon fuel, such as diesel fuel or natural gas, would result in emissions only of carbon dioxide (CO_2), water (H_2O), and nitrogen (N_2). Such combustion would involve exactly the right proportions of fuel and oxygen, burning completely at exactly the right temperature.

However, combustion in any engine is seldom ideal; the proportions of fuel and oxygen may not be correct throughout the combustion chamber; oxidation may not be complete, and so pollutants are produced. Diesel fuel may also contain impurities, especially sulphur. Thus the emissions identified as harmful to the environment and people are not so easy to eliminate in practice.

Carbon Monoxide (CO) is toxic to humans, heavier than air, and is generally a result of what might be termed an unenthusiastic combustion process, possibly from too lean a mixture; more complete or "hotter" combustion would oxidize the carbon monoxide and result in harmless CO_2.

Unburned hydrocarbons (HC) are just that; they have not been fully oxidized, either because they did not come in contact with enough oxygen, or were not subjected to sufficient heat or compression to light off the combustion reaction. While diesel engines inherently operate on a lean mixture, HC can result from uneven distribution of the fuel, and thus locally rich areas in the combustion chamber. Some unburned hydrocarbons, reacting with NOx and moisture in sunlight, can result in smog or acid rain. Unburned methane is non-reactive in this sense, but the law (except in California) and most test instrumentation does not distinguish it from reactive HC.

Oxides of nitrogen, however, do not result from incomplete combustion. The nitrogen in the air would normally be unaffected by combustion, and would be exhausted as uncombined nitrogen. If there is sufficient heat, however, oxidation takes place and the NOx component of smog and acid rain is created.

Particulates also result from incomplete combustion, but in a different way. Soot (mostly unoxidized carbon) from the diesel fuel, and ash from burnt lubricating oil form microscopic particles during the combustion process. These particles form nucleii around which other combustion by-products, such as sulphates (from diesel fuel impurities) and unburned hydocarbons can form. These still microscopic particles carry carcinogenic materials to the human lung.

The diesel combustion process

The diesel engine designer today is thus faced with the problem of controlling combustion over a wide range of speed and load conditions. The principles of diesel combustion do not make this easy. Without spark plugs, diesel engines depend on a particular property of diesel fuel: that it will oxidize spontaneously when subjected to sufficient heat and pressure provided, of course, that there is sufficient oxygen present. Thus diesel engines have high compression ratios (ranging from 14:1 to 22:1) and control the extent and timing of combustion (and the power produced) by timing the injection of a suitable quantity of diesel fuel into a combustion chamber already filled with compressed air. But even with recent improvements in fuel injection systems, combustion even at peak loads will be localized and will not occur homogeneously throughout the combustion chamber.

This leads to the undesirable emissions typical of diesel engines at part load: HC and particulates, from incomplete combustion, especially near the edges of the spray; CO, from insufficient combustion heat. However, the small quantities of fuel injected result in small emission volumes.

At somewhat higher loads, a lean form of combustion may occur throughout the combustion chamber, and the temperature will be high enough to further oxidize the CO to harmless CO2. In addition, as we move towards full load, unburned HC and particulates from the diesel fuel will be reduced.

The diesel is a relatively "clean" engine for the power it produces, making efficient use of all the fuel injected into it. However, an extended period of full load operation will result in combustion chamber temperatures rising to the threshold where free nitrogen in the intake air begins to oxidize; from this point on the production of NOx rises very rapidly. If excess fuel is injected at full load, there will not be enough oxygen to ensure full combustion, and the surplus will be exhausted as unburned hydrocarbons (HC) and particulates, or smoke; we are beyond the "smoke limit".

Thus today's diesel engineers have the difficult task of steering their combustion processes between one set of undesirable emissions associated with inefficient combustion (CO, HC, and particulates) and another set of undesirable emissions associated with efficient combustion (NOx). As mentioned above, this must be managed over a very wide range of speeds and loads.

Diesel combustion control systems

All three major north american diesel manufacturers, Caterpillar, Cummins, and Detroit Diesel Allison, together with the independent fuel system manufacturer Bosch, are currently introducing electronic engine controls for diesels in an attempt to meet the new emission regulations. Typically, these microprocessor based systems sense engine speed, fuel temperature, coolant temperature, exhaust temperature, turbocharger boost, and the position of the driver's right foot, to control not only fuel quantity and timing (through electronically controlled injectors) but also transient, overload, and low efficiency conditions which might result in undesirable emissions or excessive noise. This "drive by wire" approach also minimizes the effect of inefficient driving habits and can be used to limit engine or road speed.

Natural gas in heavy duty diesel engines

The initial impetus for the use of natural gas in diesel engines has resulted from the increasing cost and scarcity of diesel fuel. Natural gas costs about half as much as diesel, and, unlike propane, will burn effectively without premature detonation in high compression heavy duty engines.

Two approaches can be used: the first is to use diesel fuel as a "pilot fuel", serving to ignite a charge of natural gas and air; the second is to abandon the diesel fuel altogether, and convert the diesel engine to a spark ignited (SI) engine running on natural gas only. BC Research has been investigating the pilot fuel approach, as described in the following paragraphs. Ontario Research Foundation has been working on the the SI strategy, summarized in a later section.

Pilot Fuel Approach

In the pilot fuel approach, natural gas is admitted into the intake air, with the quantity of both diesel fuel and natural gas controlled by a microprocessor to produce the desired engine speed, while not exceeding combustion pressures characteristic of the unconverted base diesel engine. The result is that the speed and power produced may be made identical to that developed on straight diesel.

At full load, approximately 80% substitution appears to be possible. This results in an equivalence ratio of 0.7 - 0.75, above which there is the usual dramatic increase in emissions of all types. However, it is interesting to note that NOx emissions, indicating the onset of high combustion temperatures, are higher at full load with natural gas than on straight diesel.

To combat this, B.C. Research has carried out some interesting trials using approximately 10% Exhaust Gas Recirculation (EGR) on a Caterpillar 3208 engine. This technique, extensively used on SI engines, is unusual with diesels.

The results of the emission tests conducted at B.C. Research on the Caterpillar 3208 engine have been calculated using the SAE 13 mode test weighting. This weighting cannot duplicate the EPA test cycle, since transients have not been measured, but it does provide a rough indication of probable results. The results are:

	Diesel only	**Diesel/natural gas**
NOx (g/hp hr)	8.3	6.7
HC (g/hp hr)	0.9	13.6 (Note 1)
CO (g/hp hr)	4.3	16.0 (Note 2)
Particulates (g/hp hr)	0.5	< 0.1

Note 1: The HC emissions on diesel/gas are believed to be over 90% non reactive methane.
Note 2: The CO emissions on diesel/gas are high due to much higher part load emissions.

The main effect of EGR appears to be to absorb heat from the combustion process. at full load, this has a "cooling" effect and can reduce NOx. A secondary effect of EGR is to fill part of the combustion chamber with inert products of previous combustion; this increases the equivalence ratio, improves combustion, and reduces CO and HC emissions at part load conditions.

The **B.C. Research** work was supported by **Energy, Mines, and Resources Canada**, with the engine supplied by **Caterpillar** through **Finning Tractor and Equipment Co. Ltd.** For further information contact Dr. Lorne Gettel at (604) 224-4331.

Spark Ignition Approach

At the **Ontario Research Foundation**, work has been continuing on the development of a spark ignited (SI) version of the Cummins L-10 heavy duty truck engine, running on straight natural gas. For efficient combustion, ORF is using a specially designed high turbulence combustion chamber, combined with a "lean burn" fuelling strategy; equivalence ratios are in the .65 - .70 range. Presently, a conventional ignition system is in use, but plans call for a revised solid state ignition, with a separate low tension circuit for each cylinder. Am IMCO regulator/mixer system is used.

The engine is currently configured to produce 200 hp, turbocharged, and indications are that the thermal efficiency will run quite close to diesel. No emissions figures are yet available, as the engine is still under development.

The **ORF** work is sponsored by the **Ontario Ministry of Energy** through the **Ontario Ministry of Transportation and Communications**. For further information contact Wendell Goetz at (416) 822-4111.

CGA RESEARCH PROGRAM UPDATE

Since our last report on the **Canadian Gas Association**'s natural gas vehicle research program, changes have been made in response to new technical challenges and opportunities. Here is a brief rundown:

- Project 1.4, Direct Injection Development, has been cancelled; current research at Southwest Research Institute and the Gas Research Institute in the U.S. indicates that the injection pressure required may be too high to generate economically on board a road vehicle
- Project 3.3, Advanced On-Board Storage Cylinder Development, has been greatly expanded, with new work added on acoustic emission non-destructive testing of cylinders. The project will be managed by **B.C. Hydro Research**
- Project 2.4, Second Generation Conversion Kit Improvement, now includes work to be acrried out by **Canadian Air Sensors Inc.** on a hot wire anemometer based EFI/NG system
- Project 4.5, Diesel Engine Durability Testing, will now include both the testing of a spark ignition Cummins L-10 at **Ontario Research Foundation,** and of a pilot fuel Caterpillar 3406B at **B.C. Research**
- Project 4.6, Conversion Kit Testing, now includes both testing of the **Cryogas Engineering Ltd.** Cummins 855 pilot fuel conversion at **B.C. Research,** and of the **Air Sensors** EFI/NG kit.
- two new projects, 2.7: Programmable Ignition Timer Development and 3.5: Residential Appliance Reliability and Durability Testing, will be reported on in our next issue.

For further information, telephone John Heenan at the **Canadian Gas Association** NGV Research and Development Program, (416) 897-1733.

FIBA CANNING TESTS CHEV 454 NATURAL GAS ENGINE FOR USE IN BUSES

A 7.5 litre (454 c.i.) heavy duty Chevrolet V8 spark ignition natural gas engine is currently being tested by **FIBA Canning Inc.** for use in buses. The engine, which adapts easily to the Allison automatic transmission used for many years in GM buses, is turbocharged, uses a TNO electronic fuel/air control system, and a three way catalyst to achieve 237 hp at 3000 rpm, with minimal emissions.

FIBA Canning has had a similar engine under field test in a **Mississauga** (Ontario) **Transit** bus for some time. Drivers report excellent performance, and the engine is much smoother and quieter than the GM two stroke diesel that it replaced.

The installation and setup of the TNO electronic fuel management system, which utilizes closed loop feedback from an oxygen sensor in the exhaust pipe, was carried out at the **Ontario Research Foundation** by TNO and ORF technicians. The driving cycle used for emission testing was similar to the EPA bus cycle with the exception that "motoring" sections of the cycle were replaced by operation at a "high idle" setting. The results were encouraging:

NOx .61 g./hp hr
HC .31 g./hp hr
CO 3.95 g./hp hr

Particulate emissions were negligible, under .05 g./hp hr.

To confirm these readings and to obtain EPA certification of the engine, it has now been shipped to the EPA test facilities in Ann Arbor, Michigan. EPA certification would clear the way for test installations in U.S. buses. The **Brooklyn Union Gas Company** has given strong support to this work.

The engine development work has been supported by the **Ontario Ministry of Energy** and the **Ontario Ministry of Transportation and Communications.** For further information, contact Hugh Canning at **FIBA Canning Inc.**, (416) 299-1142.

APPENDIX 17

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Characteristics of Diesel Engines Converted to Spark Ignition Operation Fuelled with Natural Gas

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ABSTRACT

This paper reports work which is part of continuing research into the use of natural gas as an alternative fuel for automotive engines. Much work has previously been reported on the use of natural gas in small spark ignition engines and the optimisation of these engines. Since a large potential use of natural gas is in heavy transport engines and since few large spark ignition engines suitable for gaseous fuel are available from manufacturers, there is a need to investigate the conversion of existing diesel engines to spark ignition operation.

Two naturally aspirated diesel engines were characterised in the diesel mode to determine power, fuel consumption, peak combustion pressure, exhaust gas temperature and selected emissions characteristics over the engine speed and load range. After conversion to spark ignition operation these characteristics were again determined. The effects of various engine and tune parameters on performance were evaluated. The principal parameters considered were compression ratio, spark timing and air-fuel ratio.

Compression ratios of 18:1, 15:1 and 13:1 were used. This series of compression ratios permitted the effect on peak combustion pressure, exhaust gas temperature and fuel consumption to be investigated. At each compression ratio the effects of spark timing and air-fuel ratio were determined. The results show no unexpected trends compared with normal spark ignition engine operation. It is significant, however, that with appropriate tuning the natural gas engine can supply higher power than the original diesel engine. In addition, at wide open throttle, the spark ignition engines give efficiency results which are approximately equal to those from the diesel engines at similar load and speed conditions.

One of the converted engines has been optimised and installed in an urban bus which has been in operation for a year (58,000 km) with a compression ratio of 15:1. Operating experience and on-road fuel consumption data for this bus, compared with equivalent diesel powered buses are presented.

THE FIRST QUESTION TO ADDRESS is why convert a working diesel to spark ignition cycle (SI) natural gas operation. There are two major points to consider here. The first is the need to investigate alternative fuels. At present, there is little pressure to conserve liquid fuels as there is an ample world supply. This does not, however, mean that the potential for supply problems in the near future does not exist. In New Zealand, there are two main sources of liquid fuels, import and a synthetic fuel plant which uses natural gas as a raw material. A recent breakdown at that plant caused the production capacity to be cut in half which should serve to illustrate the fragility of this source. Additionally there is a very high energy cost in the conversion of methane to liquid fuels, the overall energy efficiency is about 30%. As for imports, the New Zealand Ministry of Energy estimates that there will be cheap imported oil available through to the mid-1990's. This is a very short time from now in which to investigate and develop large scale alternative fuel sources. There is also concern over the balance of supply with regard to middle distillates of which a large fraction goes to diesel fuel. In a country with a large supply of natural gas such as New Zealand, it is certainly warranted to investigate its potential as an alternative fuel in all areas.

The second major point to consider in the conversion of diesels to spark ignition is the efficiency of fuel use in the engine. This conversion may appear to be an irresponsible thermodynamic action as it is well known that diesels are more efficient than spark ignition engines. However, experimental results from both dynamometer and road testing have shown that there is not necessarily an efficiency

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penalty for the conversion. This of course depends on the duty cycle for the engine, but in this programme, 58,000 km of bus fleet operation have shown virtually equal efficiencies for a 6.4 litre direct injection diesel converted to spark ignition natural gas use when compared to diesels running similar routes.

When looking at ideal cycle efficiencies, the Otto cycle is always more efficient than the diesel cycle at equal compression ratios. It is only when we take into account throttling losses and the knock limitation on compression ratios in petrol engines that the compression ignition engine becomes more attractive.

Natural gas has an octane rating of about 120 to 130 depending on the local composition of the gas. As a result, with natural gas use, the restrictive limit on compression ratios encountered with petrol engines is eased considerably. Most modern direct injection diesels have a compression ratio of between 16:1 and 18:1. SI conversions have been run successfully at compression ratios in this range; however, the margin from knock limit is small and minor changes in fuel air ratio, inlet air temperature or ignition timing can result in knocking.

It should be noted that the optimum for compression ratio may be lower than 18:1. Many different works suggest an optimal compression ratio of about 15:1, although going below 13:1 will certainly result in a loss of efficiency.

The main disadvantage of using natural gas is that the fuel is normally stored in a gaseous state. This imposes a limit on storage capacity and consequently vehicle range. The storage tanks occupy more space and add weight to the vehicle. Although these disadvantages can be offset in many applications, they can also, in some instances, outweigh the fuel cost savings. The introduction of economic liquified natural gas installations could help make natural gas more attractive in these situations.

The main purpose of this paper is to clarify some of the issues involved in the dedicated conversion of diesels to spark ignition rather than a detailed look at the advantages and disadvantages of such a conversion. Whilst the use of natural gas in the dual-fuel mode (using a reduced quantity of diesel fuel as a pilot ignition source for the main gas-air charge) has been extensively researched, the conversion of diesel engines to operate on the spark ignition cycle is less well researched. There is a long history of manufacturers supplying SI engines to operate on gas, and some of the successful engines have had a diesel ancestry (1)*. However, the idea of a retrofit conversion to an existing diesel engine for operation in the SI mode is less well established. The final decision on whether to do the conversion and the choice between dual-fuel or dedicated conversion will depend on a number of local economic factors.

TECHNICAL ASPECTS OF CONVERSION

The most important factor in the conversi is the choice of compression ratio. Natural gas fuelled engines are able to operate at higher compression ratios compared to petrol; however, this choice must be a balance between efficiency, peak combustion pressures, exhaust gas temperatures and the possibility of engine damaging knock. Spark timing and equivalence ratio also affect these factors so that the final choice of compression ratio is the result of a compromise.

Figure 1 shows how the ideal Otto cycle efficiency (η_O) increases with compression rat (2). Also presented in this figure is the efficiency of the limited pressure or mixed cycle (η_{LP}). This cycle, which is a mixture of the Otto and diesel cycles, is used to approximate the compression ignition cycle. It is also a better approximation of the spark ignition engine cycle where combustion takes pla over a significant range of crank angle degree rather than at constant volume. In this plot the maximum pressure for the cycle is 80 times the inlet pressure. It can be seen here that the Otto cycle efficiency is always greater than the limited pressure cycle efficiency. It is also shown that the limited pressure cycle efficiency is relatively independent of compression ratio beyond a certain point, here about 13:1. The major effect of compression ratio on this cycle is a sharp reduction in exhaust temperature with increasing compressio ratio. Figure 1 shows a plot of T_4/T_1 (the ratio of end-of-expansion temperature to start-of-compression temperature) for the limited pressure cycle against compression ratio. Thi shows a reduction of end-of-expansion temperature by approximately 30% in changing the compression ratio from 13:1 to 18:1.

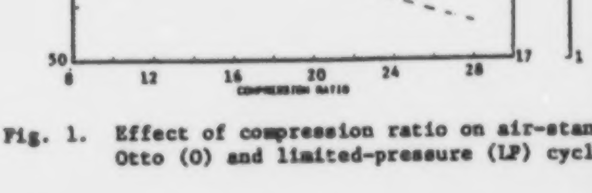

Fig. 1. Effect of compression ratio on air-standa Otto (O) and limited-pressure (LP) cycle

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*Numbers in parentheses designate references at end of paper.

Experimental data of Caris and Nelson (3) indicate an optimum efficiency for their spark ignition engine between 14:1 and 18:1 compression ratio. This range for the optimum is also supported by measurements on a single cylinder natural gas fuelled engine (4).

Based on air standard calculations, a natural gas fuelled engine running at a stoichiometric equivalence ratio is capable of about 12% greater power than the diesel engine. This is because the diesel is normally limited to an equivalence ratio of about 0.8 due to exhaust smoke considerations. This calculation also takes into account the reduced volume of intake air in the spark ignition engine due to displacement by the gaseous fuel.

The susceptibility of a gas fuelled engine to knock is very strongly dependent on compression ratio, intake temperature and equivalence ratio (5,6). Work by Karim and Klat (6) on the knock limits of methane in a CFR engine shows that with a stoichiometric mixture, an intake air temperature of 30°C gives a knock limited compression ratio of 16:1, while 100°C and similar air temperature gives a knock limited compression ratio of 13:1. These and similar data are of course engine specific, but do serve to show trends.

ENGINE TEST PROCEDURES

In the past six years there has been a variety of diesel engines converted to spark ignition in New Zealand. Table 1† lists some of these conversions. Engines number 1 and 2 are the subject of the present work, with engine number 2, a direct injection engine, being run extensively in controlled tests at several different compression ratios. Engine number 1, a 6.4 litre direct injection engine has been run both on a dynamometer and in bus fleet operation. Table 2 presents further details of the engine for which dynamometer data are reported here.

The dynamometer test procedure used for the engines was to first produce an engine map at selected speed and torque conditions in the normal diesel mode. The engine was then converted to gas operation at the original compression ratio of 18:1 and an engine map produced. The pistons were then removed, the bowls machined to give a compression ratio of 15:1 and another engine map obtained at that compression ratio. The compression ratio was then reduced to 13:1 and a final engine map produced.

The conversion procedure involves these basic steps:

(a) Removal of the injectors and housings and replacement with modified housing to carry spark plugs (Champion G58 10mm). Special care was taken to ensure optimum plug cooling.

(b) Machining of the piston bowls to lower the compression ratio, ensuring that the new combustion chamber shape has no potential hot spots and that it is conducive to high flame speeds.

(c) Removal of the injector pump and in its place installation of an electronic ignition system and distributor drive.

(d) Installation of a gas carburetor.

ENGINE TEST RESULTS

As a result of testing the engine at the various compression ratios in spark ignition natural gas operation, it was found that 15:1 was at or close to what would be considered an optimum compression ratio for this engine, subject to the compromises discussed previously. At the higher engine speeds and loads where most of the running is done for this type of engine, it was found that the maximum thermal efficiency dropped about one half percent when going from 18:1 to 15:1 and again another half percent in going from 15:1 to 13:1. At a compression ratio of 18:1, it was possible to encounter engine knock while at a compression ratio of 15:1 it was demonstrated that there was a good margin for knock under all operating conditions.

The results given here in Figures 2 to 7 are a comparison of operating the engine in the normal diesel mode versus the spark ignition conversion at a compression ratio of 15:1. The spark ignition engine was tuned to match the maximum power point of the diesel at 3000 rpm and produced higher wide open throttle (WOT) power at lower engine speeds.

Comparing the thermal efficiency plots for the two engines, (Figures 2 and 3), it can be seen that there is little difference along the WOT 350 Nm line. For the SI engine there is a small drop-off in efficiency relative to the diesel from peak torque down to about half torque but from there down a much steeper drop in efficiency occurs with decreasing torque due to throttling losses. This drop-off in efficiency is less marked at higher engines speeds, however, where most of the running under load is done.

Exhaust temperature as a function of torque and engine speed are shown in Figures 4 and 5. For the diesel it can be seen that the temperature falls very rapidly with load at all engine speeds, while for the SI engine there is very little change in exhaust temperature with load. It should also be noted that the exhaust temperature of the SI engine is about 60°C higher at peak torques than for the diesel at comparable speeds, though the SI is producing higher torque than the diesel at lower speeds.

Figure 6 shows the peak cylinder pressure for the diesel engine which has a maximum of about 80 bars with only a small drop in pressure with decreasing torque. Figure 7 shows that for the spark ignition conversion, at corresponding torques and speeds, the peak pressures are slightly lower at high torques with a large reduction in peak pressure at lower torques compared with the diesel. Along the top of the torque curve at lower engine speeds the peak pressures are higher for the SI

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†See end of paper for Tables.

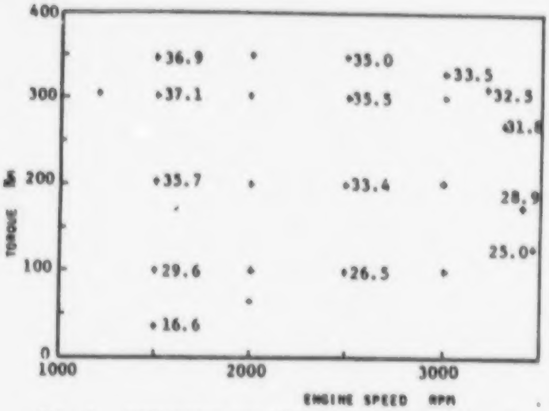

Fig. 2. Efficiency map (%) for 5.6 litre diesel engine.

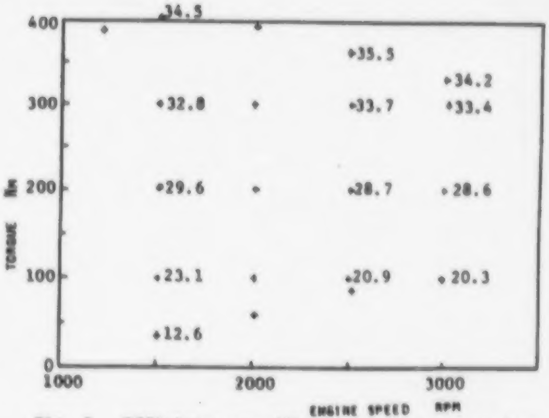

Fig. 3. Efficiency map (%) for engine converted to SI operation; CR = 15:1, ignition timing 15° BTDC.

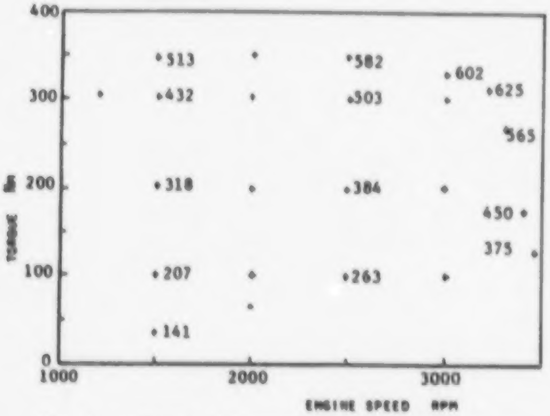

Fig. 4. Average exhaust temperature map (°C) for original diesel engine.

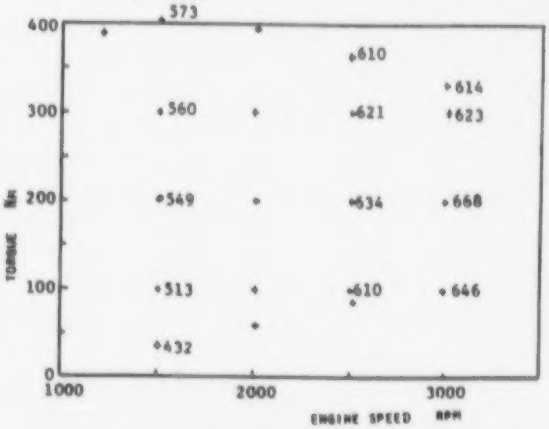

Fig. 5. Average exhaust temperature map (°C) for SI conversion: CR = 15:1, ignition timing 15° BTDC.

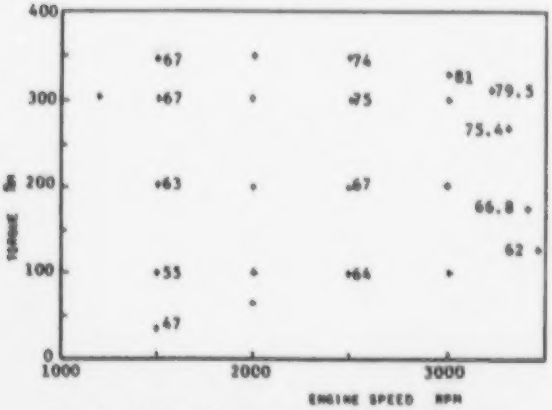

Fig. 6. Peak cylinder pressure map (bar) for original diesel engine.

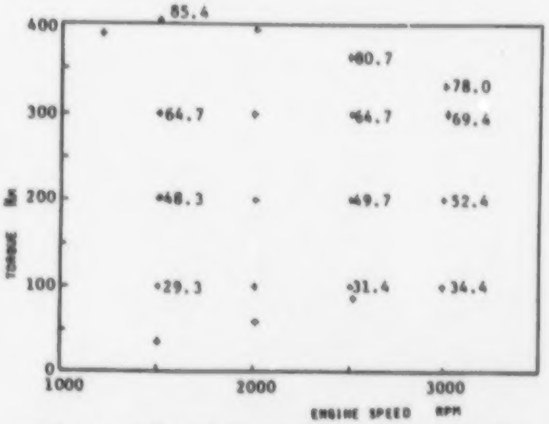

Fig. 7. Peak cylinder pressure map (bar) for SI conversion: CR = 15:1; ignition timing 15° BTDC.

than for the diesel, but the power produced at these points is correspondingly higher. At 3000 rpm in the spark ignition engine, the peak pressures fell from 88 to 78 to 71 bars with a drop in compression ratio from 18:1 to 15:1 to 13:1.

The spark ignition engine was tuned to match the diesel at peak power at 3000 rpm. For the 15:1 compression ratio this corresponded to an equivalence ratio of 0.98. Although the carburettor used richened the mixture somewhat when reducing torque or speed, the equivalence ratio throughout the range remained between 0.98 and 1.12. The ignition timing was set at a point slightly away from the maximum efficiency and torque point. Figure 8 shows a plot of efficiency, torque, peak pressure and exhaust temperature versus ignition timing at 15:1 compression ratio and 1500 rpm. Similar results were obtained at 3000 rpm. It can be seen here that the timing for maximum torque and efficiency is at 8°BTDC. The timing was set at 15°BTDC because of the exhaust gas temperature considerations. It is shown here how efficiency falls with timing advance but there was only a small penalty in efficiency for doing this and a significant improvement in exhaust temperature.

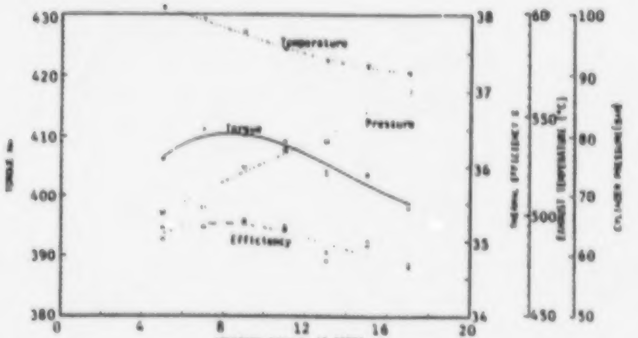

Fig. 8. Plot of torque, thermal efficiency, peak cylinder pressure and exhaust temperature against ignition timing; 1500 rpm, WOT.

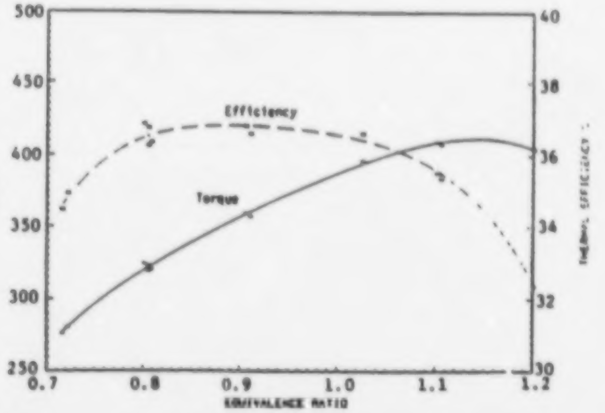

Fig. 9. SI engine torque and thermal efficiency against equivalence ratio; 1500 rpm, WOT, ignition timing 9° BTD.

Figure 9 illustrates the variation of efficiency and torque with equivalence ratio. The equivalence ratio was adjusted between 0.7 and 1.2 by varying the gas flow at WOT. The torque here peaks at an equivalence ratio of 1.12. The efficiency increases with decrease in equivalence ratio down to about 0.8 and from there it falls off rapidly due to lean misfiring.

Data similar to those presented in Figures 8 and 9 were also obtained at compression ratios of 18:1 and 13:1 and exhibit similar characteristics. The 'optimum' ignition timings determined for the different compression ratios were: 7°, 15° and 17°BTDC for compression ratios of 18:1, 15:1 and 13:1 respectively.

ON ROAD EXPERIENCE

A programme was funded by the Liquid Fuels Trust Board in New Zealand to carry out road trials with a spark ignition conversion of a diesel engine. An engine conversion was done (Table 2, item 1), run on the dynamometer and installed in a bus. That bus has now been operated for 58,000 km over the last year with close monitoring. The converted engine was a 6.4 litre direct injection diesel and the compression ratio was reduced to 15:1 by machining the piston bowls. Table 3 presents the engine specifications for the original diesel engine and converted SI engine. Original plans to test other compression ratios were dropped because there were no problems with knock encountered during operation at 15:1. The bus was fitted with six 80 litre storage cylinders with two fill points for refuelling. The refuelling time for the bus was approximately 8 to 10 minutes. The bus was part of a fleet run by an Auckland bus company and the route was a mixed city/urban route with some long steep hills.

Over a 9-month period the average fuel consumption of the bus was 1095 MJ/100 km (25.5 kg/100 km). This compares with 1040 MJ/100 km (28.8 ℓ/100 km), which is an average for two different diesel buses running similar routes. Monitoring over different time periods showed that the gas bus varied from 5% lower to 5% higher on fuel consumption compared with the diesel buses. The reason that the efficiencies are nearly the same for the two engines can be explained by looking at measured driving patterns. Although the relative efficiency of the spark ignition engine drops significantly below that of the diesel engine at lower torques, most of the fuel is used at higher loads where the efficiencies are virtually the same. Measurements show that 75% of the fuel is burned in 35% of the time while the engine is running at greater than 60% of maximum torque. For 50% of the time, the engine is running at lower than 20% of maximum torque, however, this only accounts for about 15% of the total fuel consumed.

As for maintenance, there were only two areas where the gas bus required more attention than the diesel and these were for spark plugs and exhaust valve tappet clearances. The spark plugs needed to be gapped every 10,000 km and replaced every 20,000 km. Although inlet valves were seemingly unaffected, exhaust valve tappet clearances needed to be adjusted more often than for diesel operation. Apparently the higher exhaust gas temperatures caused some recession of the valve seats. The engine oil was monitored closely during these trials and standard diesel engine oil was found to be satisfactory for use with the natural gas fuelled engines.

One additional benefit of running on natural gas was a reduction in noise level. At 80 km/hr road speed interior sound levels were reduced 2.5 dBA from 79 to 76.5 and curbside measurements showed a 5 dBA reduction from 88.5 to 83.5.

Driver response to the gas bus was somewhat mixed; they liked the reduction in noise levels although they were used to driving by engine noise and had to readjust their driving habits to the quieter engine. Although dynamometer tests and time trials up hills have shown the gas engine bus to have equal or greater power compared to the diesel, the drivers still felt gas engines to be less powerful than the diesels. This was apparently due to the difference in throttle response and the reduced noise levels.

The original converted engine has been removed from the bus and will be returned to the manufacturer in Japan for a close inspection. A second engine has been converted and is currently being run in further trials.

CONCLUSIONS

This experimental programme has shown that it is technically feasible to convert a direct injection diesel engine to a high compression ratio spark ignition engine and to obtain comparable on-road efficiency from the two engines. Careful dynamometer optimisation was required to achieve this result and in order to avoid excessive exhaust temperatures and combustion pressures and to avoid engine knock.

ACKNOWLEDGEMENTS

The financial support of the Liquid Fuels Trust Board for the experimental programme reported here is gratefully acknowledged.

The assistance of Dr. J. McFeaters in final paper preparation is also gratefully acknowledged.

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Table 1: Spark Ignition Natural Gas Conversions of Diesel Engines in New Zealand

Item	Engine Type	Supporting Organisation	Test Organisation	End Use
1	6.4 litre DI	Liquid Fuels Trust Board	Vehicle and Fuels Research Unit	Bus Trial
2	5.7 litre DI	Liquid Fuels Trust Board	Vehicle and Fuels Research Unit	-
3	5.7 litre DI turbocharged	Liquid Fuels Trust Board	Vehicle and Fuels Research Unit	-
4	6.4 litre DI	Palmerston North City Corporation	TransGas Services Limited	Bus Fleet Operation
5	6.2 litre DI	Greyhound Buses	TransGas Services Limited	Bus Fleet Operation
6	6.22 litre DI	-	Transport Fuel Systems (NZ) Limited	Agricultural Tractor
7	14 litre DI turbocharged	Carter Holt Harvey/ Liquid Fuels Trust Board	Transport Fuel Systems (NZ) Limited	Short Haul Logging Truck
8	5.7 litre DI	Ministry of Works and Development	Transport Fuel Systems (NZ) Limited	General Fleet Use
9	5.3 litre DI	Ministry of Works and Development	Transport Fuel Systems (NZ) Limited	General Fleet Use

Table 2: Diesel Engine Specifications

Type:	Direct injection, 6 cylinder, in-line, 4 stroke
Capacity:	5.654 litres
Compression ratio:	18:1
Maximum rated output:	150 hp at 3000 rpm
Aspiration:	Natural

Table 3: Engine Specifications for Road Trials

Diesel-original configuration

Type:	6 cylinder, in-line, 4-stroke
Capacity:	6.44 litres
Compression ratio:	17.9:1
Rated power:	106 KW at 2800 RPM

Spark Ignition Conversion

Compression ratio:	15:1
Spark timing:	Static 15° BTDC Centrifugal advance nil Vacuum advance 20° below 20 kPa manifold vacuum

APPENDIX 18

RESEARCH SUMMARY

Title: Gas Engine Durability Improvement
Phase I - Information Gathering

Contractor: Southwest Research Institute
GRI Contract Number: 5085-293-1179

Principal Investigator: M. B. Treuhaft

Report Period: October 1985 - June 1986
Topical Report

Objective: To develop a solution to valve and valve seat durability problems which are inherent in natural gas-fueled, four-stroke cycle, spark-ignited engines. A further objective is to determine the feasibility of substantially extending the normal service interval for gas-fueled engines.

Technical Perspective: This program addresses a need to improve the cost effectiveness of natural gas-fueled engines by significantly reducing the maintenance costs. This improvement can be accomplished by developing an understanding of the causes of valve and valve seat wear and finding a solution to this problem. It is also necessary to analyze other maintenance items requiring periodic attention and to evaluate methods to increase the time between service intervals. Economic benefits will accrue to natural gas-fueled engine users through reduced maintenance costs, higher utilization of capital equipment, and a reduction in capital expenditures for standby engines or engine replacements. Engine manufacturers as well as valve train component manufacturers will benefit through an upgrading of valve train design and materials technology as applicable to natural gas-fueled engines.

Results: Valve/valve seat problems were found to exist. In some cases they are severe; in others they are less noticeable because maintenance schedules have been tailored to make the necessary adjustments and repairs before major failures occur.

Maintenance practices vary widely and usually do not follow manufacturers' recommendations.

Users express concern regarding crankcase lubricants. Some experiment with lubricants until they find the best lubricant for each engine model. Others express a desire to have more guidance regarding which lubricants should be

APPENDIX 19

Performance and Emissions of Propane, Natural Gas, and Methanol Fuelled Bus Engines

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ABSTRACT

A comparative evaluation of six transit bus engines (three diesel, one propane (LPG), one natural gas for vehicles (NGV), and one methanol) has been performed. The purpose of the program was to assess the exhaust emissions and fuel consumption of current state-of-the-art large alternative fuel engines.

Engine dynamometer test work was performed at the Ontario Research Foundation (ORF) which allowed a detailed comparison of several alternative-fuelled engines versus their diesel counterparts. Test data includes steady-state brake-specific fuel consumption maps, torque and horsepower curves. Transient performance, fuel consumption and emissions information came from computer-controlled engine dynamometer runs of the Advanced Design Bus (ADB) test cycle as defined by US DOT/Booz, Allen & Hamilton (1)[*].

Engine speed and load requirements were determined from data supplied by Detroit Diesel Allison (DDA) for their 40 foot urban transit bus with Canadian DDA 6V71 NA (C55) engine. This information was in the form of a computer printout which listed engine speed, load, transmission gear, vehicle acceleration and position as a function of time. These data were fitted to the ADB cycle requirements at the Ontario Ministry of Transportation (MOT) in support of this program. A listing of the engine speed and torque versus time for all phases of the ADB cycle was provided to ORF where it was implemented on a computer-controlled engine dynamometer.

The results of the test program indicate that alternative fuels have a useful role to play in reducing transit bus engine exhaust emissions.

THE CURRENT PRACTICE in the North American urban transit industry is the use of diesel fuel in urban buses. Of the few diesel engines suitable for urban bus use in Canada, the DDA 6V71 NA is by far the most popular. For this reason, it is desirable to gauge the performance of any other candidate engines against the DDA baseline.

Previous investigations regarding the availability of large alternative-fuelled engines resulted in the procurement of Iveco LPG (Liquefied Petroleum Gases - propane)(2) and NGV (Natural Gas for Vehicles - compressed methane gas) engines for ATF (Alternative Transportation Fuel) urban bus demonstrations. The corresponding Iveco diesel engine was also obtained for baseline comparison purposes.

The MAN methanol engine was obtained along with its diesel counterpart for similar comparison work. This methanol engine had been developed to the state where it was equipped with a conventional oxidation catalyst and it was tested in that form. This engine was unique in that it employed direct cylinder injection with spark ignition (4,5) compression ignition-type cycle. The MAN methanol engine also served to highlight emission reductions possible by employing catalytic exhaust after-treatment.

The assessment of exhaust emissions was intended to determine the relative effect of alternative-fuelled bus engine use on air quality (if they were to replace existing engines). It was not intended to assess compliance with existing or future heavy-duty bus emission standards.

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[*] The numbers in parentheses refer to references found at the end of this paper.

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All of the engines tested were of four-stroke design except the DDA 6V71 NA (two-stroke).

TEST PROGRAM

The test program included two distinct phases, steady-state and transient. The steady-state tests provided engine fuel consumption data for future computer simulation work. The transient data was the basis for evaluation of exhaust emissions and engine energy efficiencies in an environment representative of that experienced by the bus engines in revenue service.

TEST FACILITIES

The facilities used to conduct the engine testing consisted of a 500 HP eddy current dynamometer in combination with an HP 1000 computer system. Flexibility in the system allowed either manual operation or, through software building, running of various transient engine operating schedules.

Engine load was controlled through an actuator operating either a throttle (homogeneous charge engines) or a governor input lever; speed was controlled through variation of the load on the dynamometer.

Type T and K thermocouples were used for monitoring of charge air, fuel, coolant lubricant and exhaust temperatures, while Viatran transducers were employed to monitor pressures at critical points in the intake manifold, exhaust, oil gallery, fuel supply line.

Air flows were monitored using a Meriam flow meter. Fuel flows were determined using either the gravimetric method (IVECO Propane), a flow-through Flowtron fuel meter (DDA 6V-71, IVECO Diesel, MAN Diesel) for liquid fuels and a Rotex flow-through meter for natural gas.

Number 1 diesel fuel was supplied from an underground tank through a lift pump. Chemical grade methanol was supplied from 200 litre drums and propane was procured in 40 lb. cylinders, which were placed on a scale. Natural gas was taken from a 40 psig building supply line, reduced to 6" H_2O through a pressure regulator and directly introduced into the mixer on the engine. All instrumentation was calibrated before testing. At steady-state operation, data sampling was repeated at a given speed load point to check for consistency in results. During transient testing, the total fuel consumption was monitored and the overall cycle efficiencies were calculated based on these data.

During transient testing, a MACSYM 150 computer system was used for collecting data and performing regression analysis for cycle validations. Data collected on a second by second basis were then transferred to an IBM PC for further detailed calculations and analysis. Variables sampled included engine speed, load, gaseous emissions (THC, CO, CO_2, NO_x) and fuel flow (for diesel fuel only).

EMISSIONS - To characterize the emissions under steady-state and transient cycle operation, a 2000 cfm PDP-CVS double dilution tunnel was used. This system was built according to EPA specifications.

A Beckman gaseous emissions bench was used in conjunction with the computer system to record real time concentration of carbon dioxide, carbon monoxide, total hydrocarbons and oxides of nitrogen. The particulate bench provided controlled proportional sampling of particulate matter. Calibration of these benches was performed regularly during testing.

STEADY-STATE ENGINE MAPPING - Each engine was mounted on the engine dynamometer and instrumented for temperatures and fuel flowrates. Data was collected at six different engine speeds and five different load settings. The data was then plotted in the form of energy efficiency observed at the above speed and load settings (See Figures 1(a), 1(b), 1(c), 1(d), 1(e), 1(f)).

ADB TRANSIENT CYCLE TESTING - After reviewing the literature regarding transient engine testing, the choice was made to employ the Advanced Design Bus (ADB) transient cycle as the most realistic and common bus cycle characterization. The empty bus configuration was chosen, as this more closely approximated the engine loading of the EPA heavy-duty engine transient emission test cycle.

In order to implement the ADB cycle on the ORF engine dynamometer, cycle requirements were compared to the GM DDA 6V71 NA (C55 injectors) performance data supplied by DDA. This included engine torque and power values versus time for a 40 foot bus continuous full-load acceleration from zero to fifty-five miles per hour (governor bumping speed). These data were then fit to the basic ADB cycle elements, the central business district (CBD), the arterial (ART) and the commuter (COMM) phases.

Each cycle element consists of a full-load acceleration period followed by a cruise period, a deceleration period and an idle period (See Figures 2(a), 2(b), 2(c)).

The CBD phase includes 14 elements of accelerations to 20 mph each followed by

FIGURE 1(a) DDA 6V71 NA (C55)

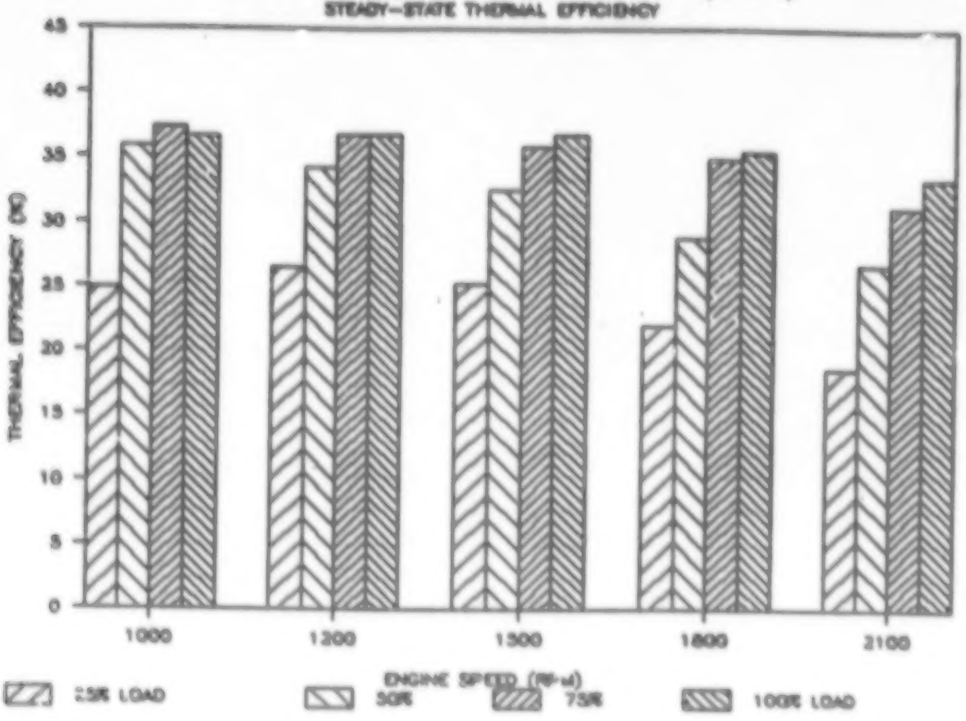


FIGURE 1(b) IVECO DIESEL

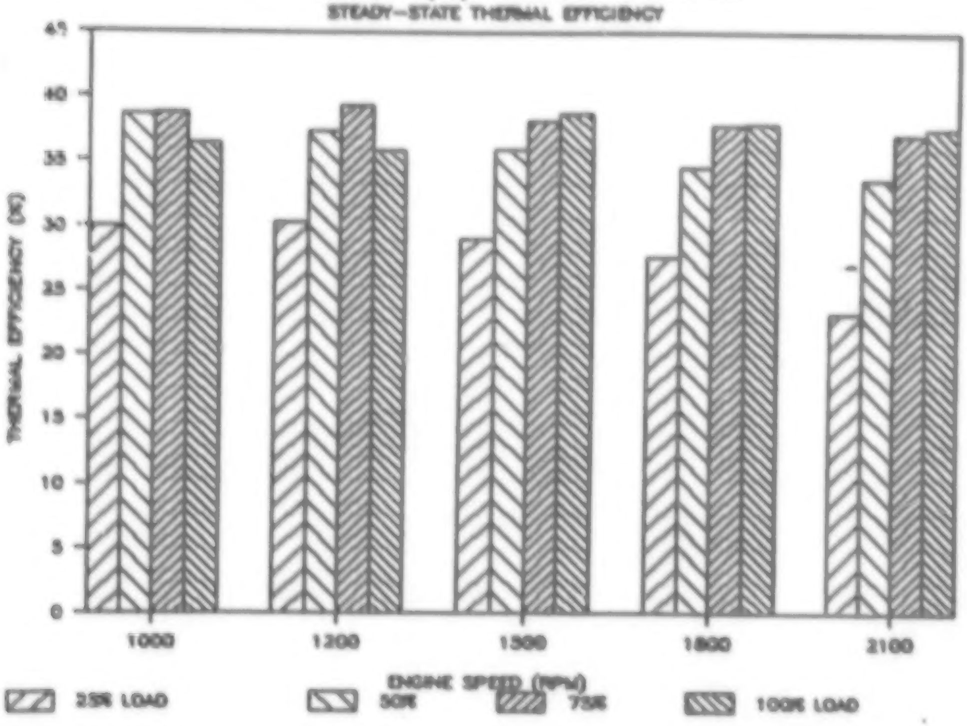


FIGURE 1(c) MAN DIESEL

STEADY-STATE THERMAL EFFICIENCY

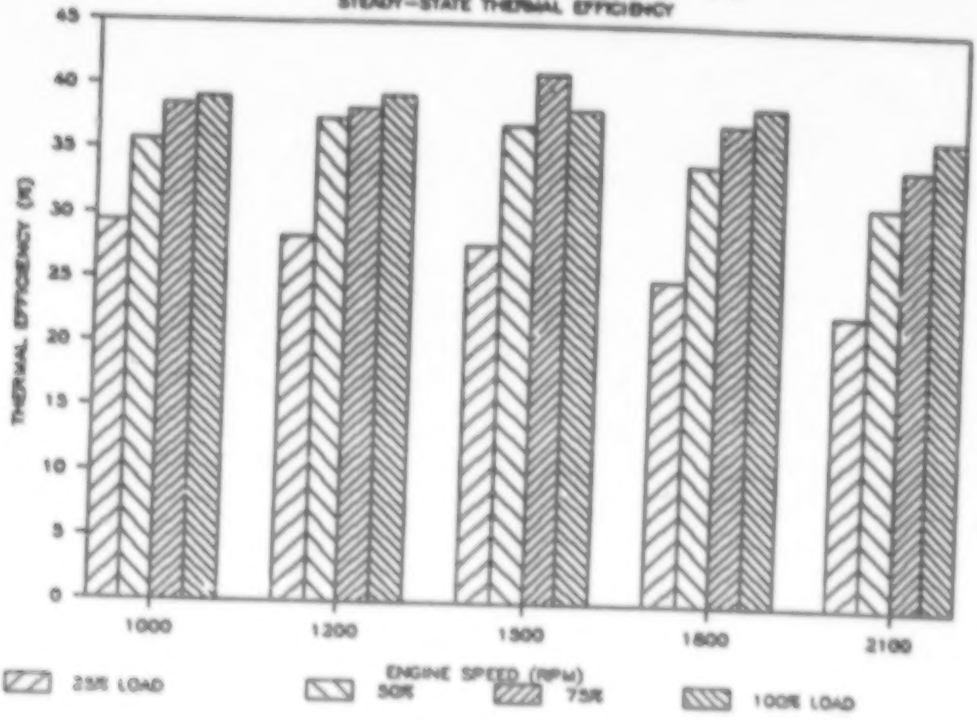


FIGURE 1(d) IVECO PROPANE (LPG)

STEADY-STATE THERMAL EFFICIENCY

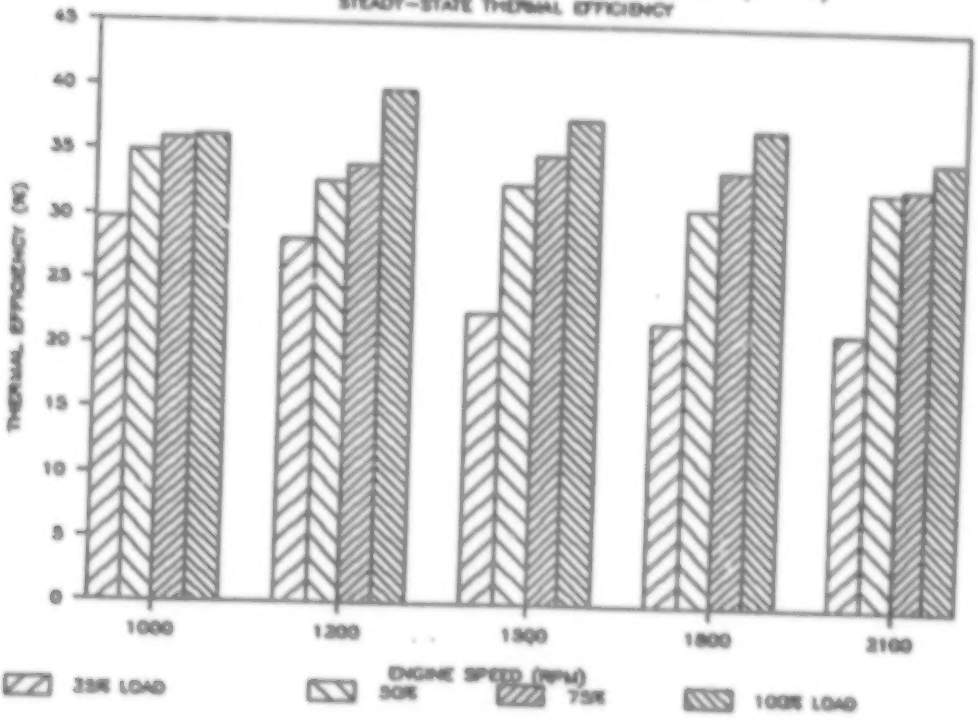


FIGURE 1(e) IVECO NATURAL GAS (METHANE)

STEADY-STATE THERMAL EFFICIENCY

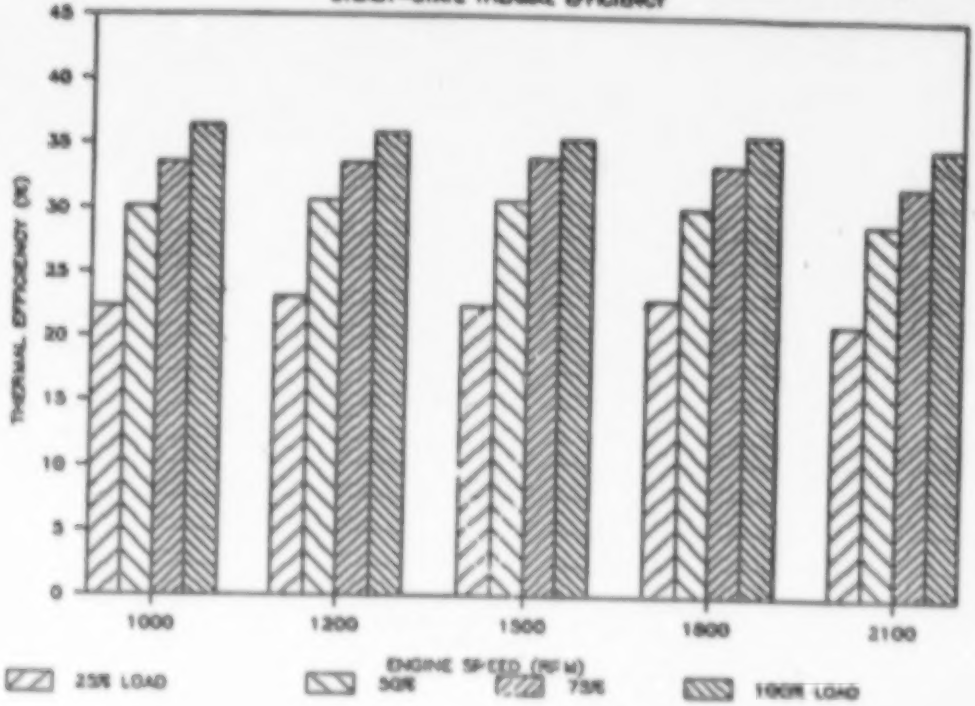


FIGURE 1(f) MAN METHANOL (M100)

STEADY-STATE THERMAL EFFICIENCY

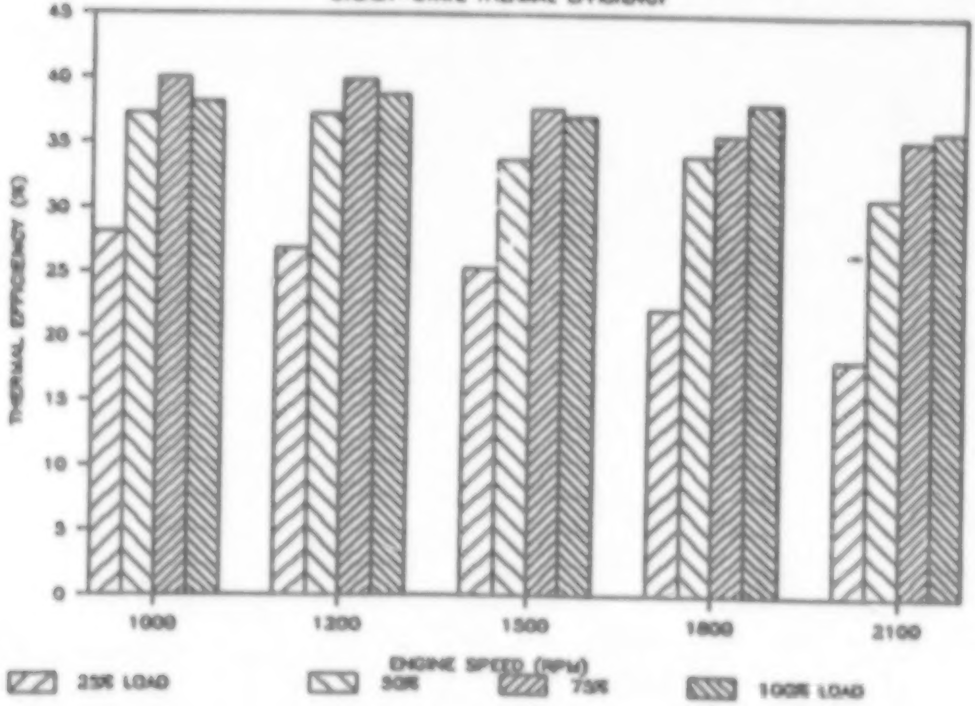


FIGURE 2(a) – ADB CYCLE (CBD)

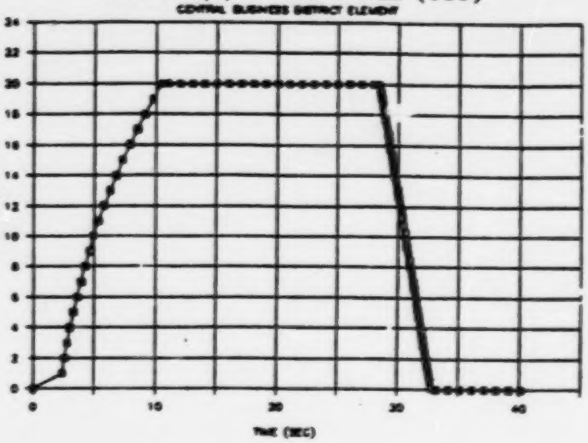


FIGURE 2(b) – ADB CYCLE (ART)

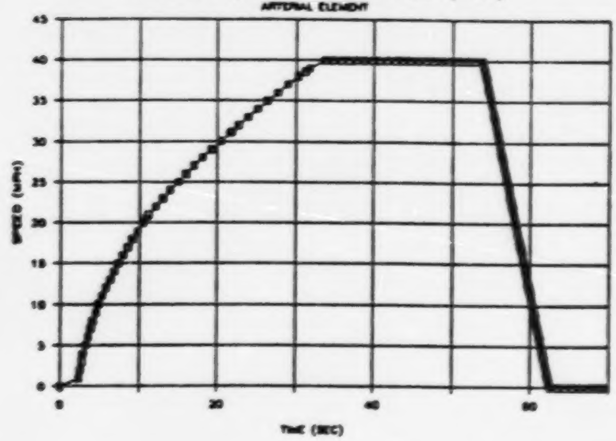
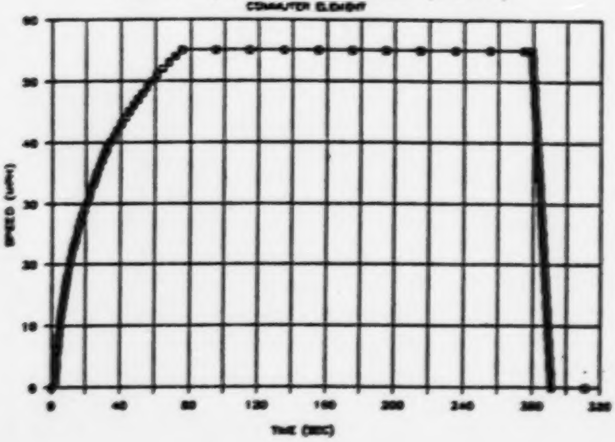


FIGURE 2(c) – ADB CYCLE (COMM)



a short cruise period and a deceleration phase at -6.78 ft/s 2 and 7 seconds of idle. The total distance covered in 14 repetitions is 2 miles.

The ART phase has 4 accelerations to 40 mph each followed by cruise, deceleration and 7 seconds of idle. A total distance of 2 miles is covered.

The COMM phase involves a single acceleration to 55 mph then cruise, deceleration and 20 seconds of idle resulting in 4 miles being covered. The complete ADB cycle consists of the components shown in Table 1.

TABLE 1

ADB TRANSIENT CYCLE PHASES

1. START
2. CBD
3. IDLE (5 MINUTES)
4. ART
5. CBD
6. ART
7. CBD
d. COMM

Total Distance = 14 miles.
Total Cycle Time = approximately 47 minutes.

FUEL PROPERTIES/ANALYSIS

The following fuels were employed in the test program:

1. Diesel (#1)
2. Propane (LPG)
3. Natural Gas for Vehicles (NGV, methane
4. Methanol (Neat, M100)
5. Fuel Methanol (M85) 85% methanol
15% super unleaded gasoline

Fuel energy values were obtained from the Handbook of Natural Gas Engineering (3) except for diesel #1 for which analysis was available (See Table 2).

TABLE 2

TEST FUEL DATA

Fuel	Energy Content (LHV)	LHV Relative to Diesel	Stoichiometric Air/Fuel Ratio (Mass Basis)
Diesel (#1)	42.5 MJ/kg	1.00	14.5*
Propane	46.1 MJ/kg	1.08	15.6
NGV	50.1 MJ/kg	1.18	17.2
M100	20.0 MJ/kg	0.47	6.4
M85	23.5 MJ/kg	0.55	7.6

* Diesel engines operate with lean air-fuel mixtures due to smoke limitations.

The propane (LPG) supply in Ontario is normally about 96% propane with the remainder being mostly butane and propylene (HD-5 Specification). NGV is about 96% methane with low water content. Fuel methanol (M85) is 85% chemical grade methanol with 15% super unleaded gasoline (seasonally blended).

ENGINE SPECIFICATIONS AND MEASUREMENTS

TABLE 3

TEST PROGRAM ENGINE DATA

MAKE	CONFIG	DISP (l)	FUEL SYS (type)	C.R.	POWER (hp) @ RPM	TORQUE (ft-lb) @ RPM	INTAKE (type)
DDA 6V71 (2-stroke) DIESEL	V6	7.0	D. INJ.	17	163 @ 2100	560 @ 1000	BLOWER
IVECO 8220. R DIESEL	I6	9.6	D. INJ.	17	174 @ 2100	488 @ 1100	NA
MAN D 2566 UH DIESEL	H6	11.4	D. INJ.	18	170 @ 2200	508 @ 1300	NA
IVECO 8220. RLPG OTTO CYCLE	I6	9.6	IMPCO CA425	10.9	174 @ 2100	502 @ 1000	NA
[illegible] IVECO 8220. RCNG OTTO CYCLE	I6	9.6	IMPCO CA425	12	178 @ 2100	489 @ 1400	NA
MAN D 2566 FMUH METHANOL SPARK-ASSISTED DIESEL CYCLE (Conventional Oxidation Catalyst (COC) equipped)	H6	11.4	D. INJ.	18	204 @ 2200	632 @ 1200	NA

H6 - Horizontal Inline 6 cylinder
D. INJ. - Direct Injection
NA - Naturally Aspirated

EXHAUST EMISSIONS TESTING - The emissions of particulates as well as gaseous pollutants (CO, HC, NOx, CO2) were measured while the engines were operating over the transient ADB cycle. This was accomplished by sampling the exhaust gases over the entire test cycle followed by chemical component analysis. Fuel flowrates were monitored as directly as possible and, in some cases, instantaneously. Particulates were collected on paper filters in the dilution tunnel and then weighed.

After initial testing, the Iveco natural gas engine was optimized for nitrogen oxide emissions, while attempting to maintain comparable power and torque output. The optimization philosophy was one of leaning the mixture under part-load conditions, while adding a load-sensitive spark advance component to the ignition curve to offset the power loss. The engine was mapped for minimum spark advance for best torque (MBT) and resulting NOx emissions over a range of engine speed, load, and air/fuel ratios. The ignition and fuel settings were then implemented in hardware. The results for the optimized version are presented along with those for the original version of the engine.

RESULTS

ENGINE TORQUE CURVES - The MAN methanol engine produced the highest torque averaging about 117% of the DDA 6V71 NA result. The DDA was next in the engine speed range of 1000 to 1600 RPM. In this same speed range, the Iveco diesel and Iveco propane engines were down to 88% of the DDA output. The Iveco natural gas engine was down to 83% at 1000 RPM but then increased and surpassed the DDA above 1600 RPM. This trend continued to the 2100 RPM level where the natural gas engine demonstrated a 115% torque level compared to the 6V71 NA. The Iveco diesel and propane engine torque outputs remained below that of the DDA until 1900 RPM was reached and then surpassed the DDA output, matching the natural gas engine torque at 2100 RPM (See Figure 3).

ADB TRANSIENT CYCLE ENERGY EFFICIENCY - Based on a calculation of engine work done versus fuel consumed during the ADB cycle, engine energy efficiencies were compared to that of the DDA 6V71 NA. The DDA two stroke diesel displayed a cycle energy efficiency of 31.1% The Iveco four stroke diesel was the most efficient at 116% of the DDA diesel followed by the MAN diesel at 112%. The MAN methanol engine was equally efficient as the DDA 6V71 on both M100 and M85 as was the Iveco propane engine. The lowest energy efficiency was displayed by the Iveco natural gas engine at 95% of the DDA for the original version and 96% of the DDA engine for the optimized (reduced NOx) version (See Figure 4).

The results show that the propane and methanol engines exhibited higher cycle efficiencies than the natural gas engine. Also of note are the higher efficiencies of the four-stroke diesel engines.

PARTICULATE EMISSIONS - All of the alternative-fuelled engines were very low in particulate emissions. The lowest particulate levels were from the catalyst-equipped MAN methanol engine. When operating on 100% neat methanol or M85, the level was approximately 6% of the DDA engine. The Iveco propane engine was next at 9% and the natural gas engine showed 16%.

The DDA 6V71 was the best of the diesels at 0.17 g/bhp-hr. The Iveco diesel and MAN diesel were at 124% and

FIGURE 3 – ENGINE TORQUE DATA

BUS ENGINE TEST PROGRAM

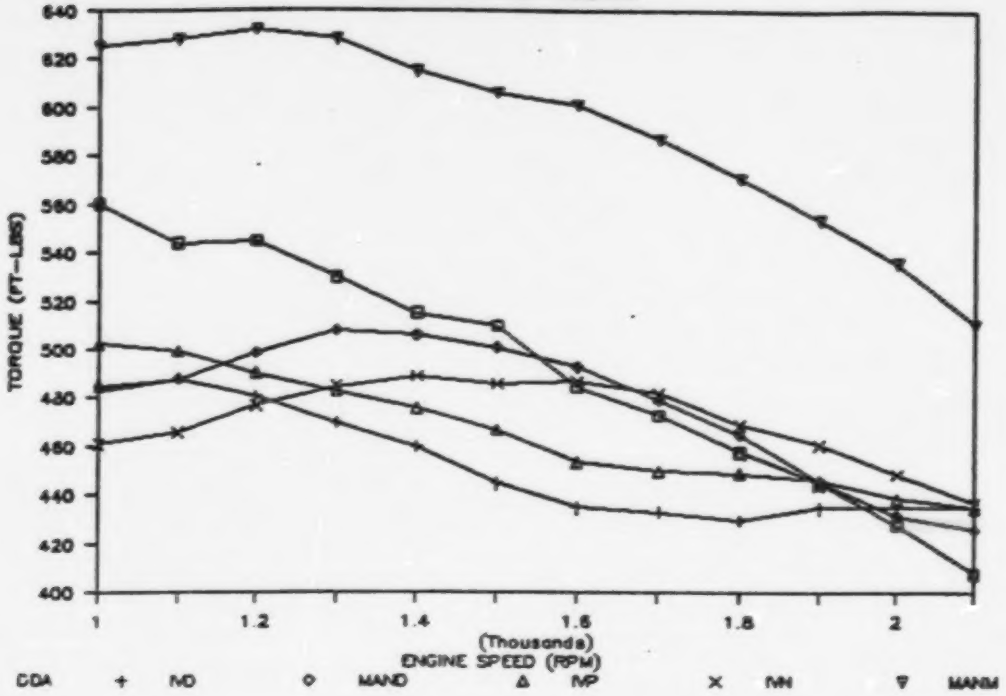


FIGURE 4 – ADB CYCLE EFFICIENCY

RATIO OF (ENGINE DATA/DDA 6V71 DATA)

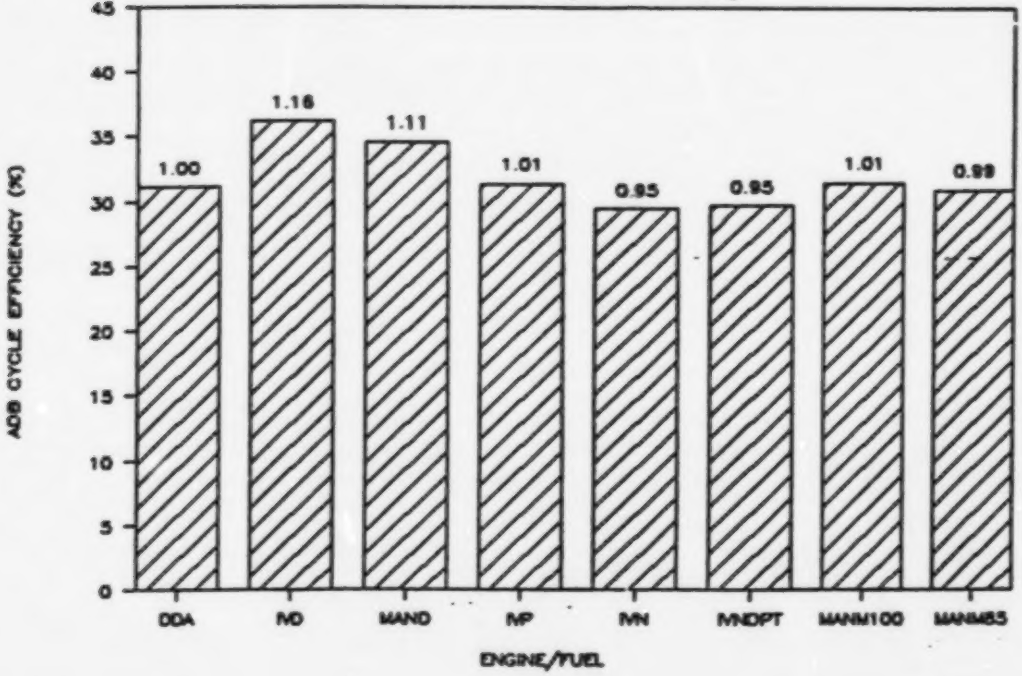
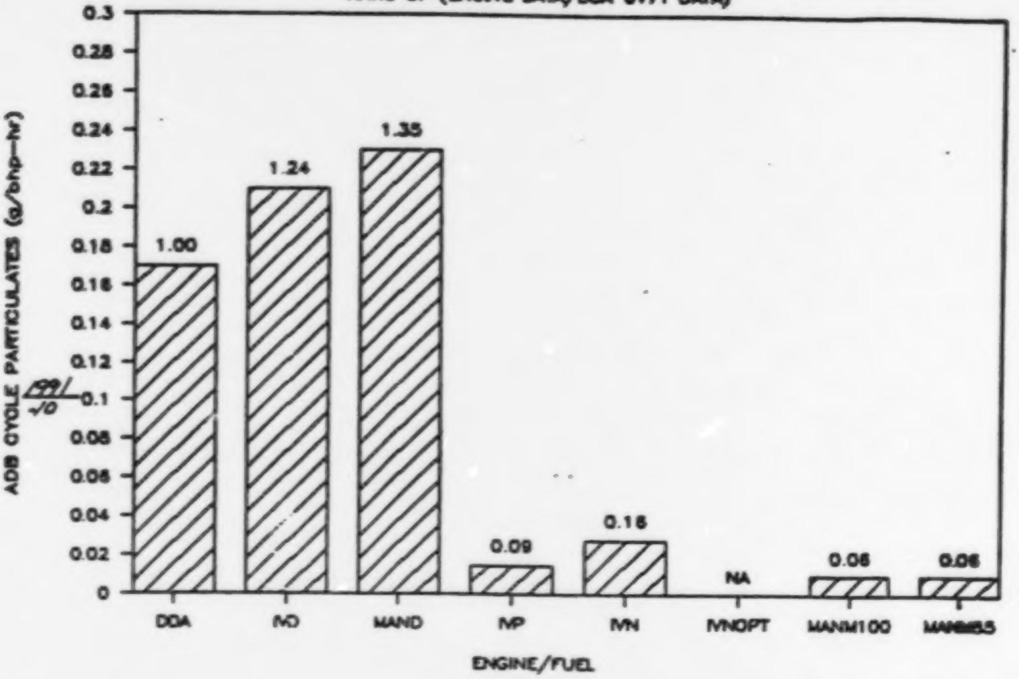


FIGURE 5 – ADB CYCLE PARTICULATES

RATIO OF (ENGINE DATA/DDA 6V71 DATA)



1991/.10

1991 15.5

FIGURE 6 – ADB CYCLE CO

RATIO OF (ENGINE DATA/DDA 6V71 DATA)

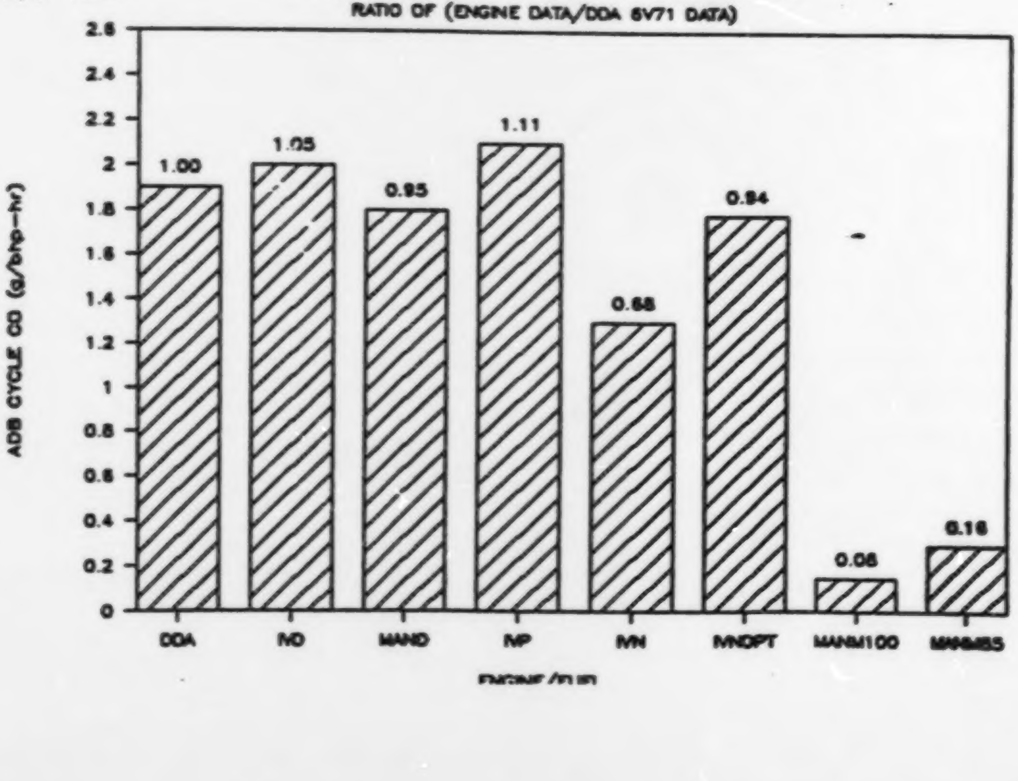


FIGURE 7 – ADB CYCLE HC

RATIO OF (ENGINE DATA/DDA 8V71 DATA)

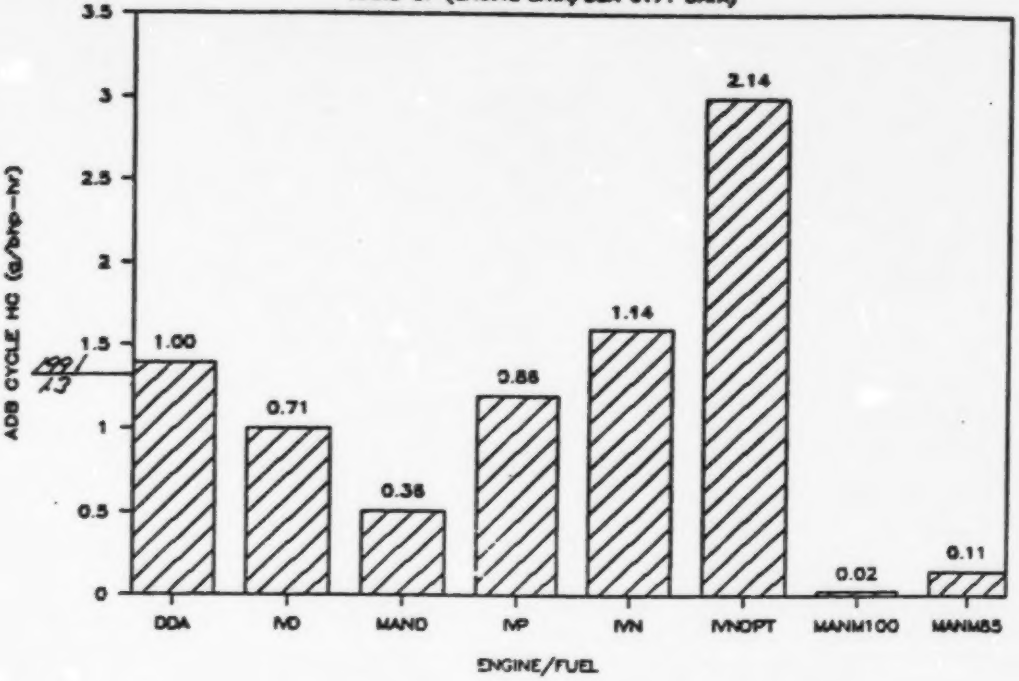
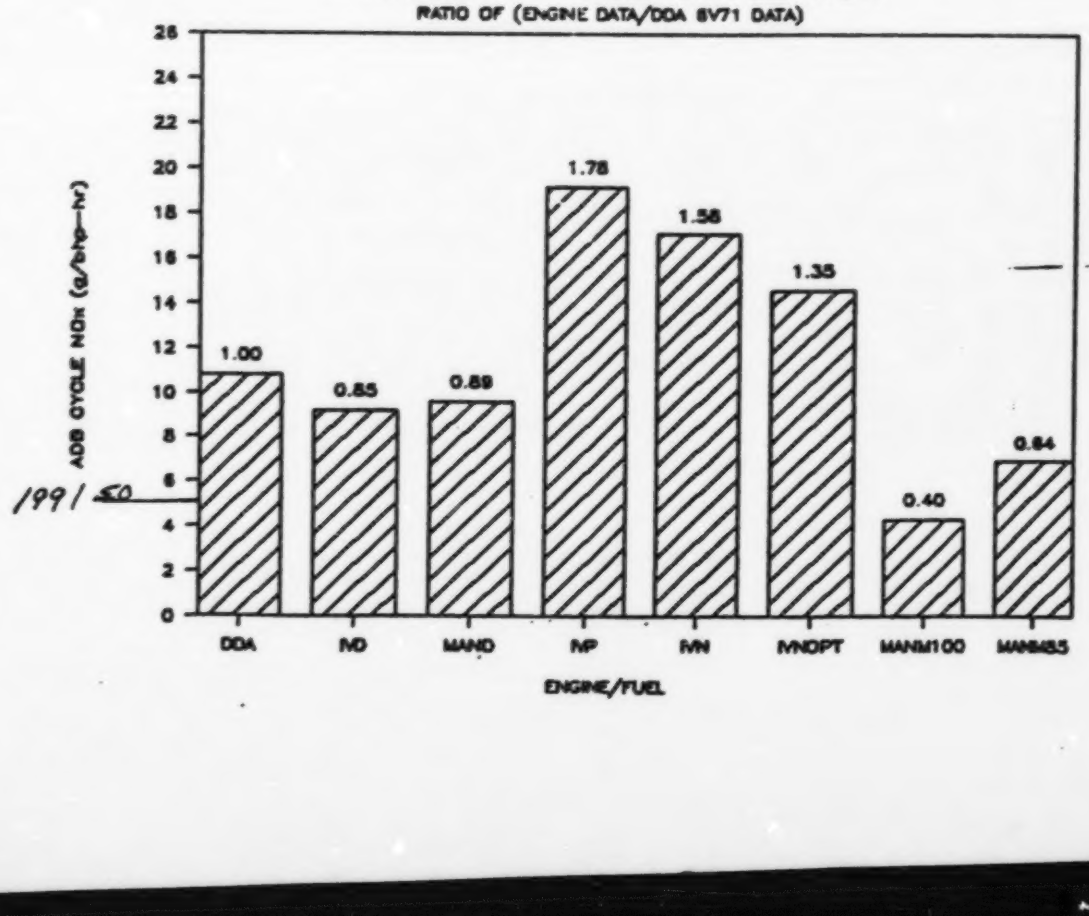


FIGURE 8 – ADB CYCLE NOx

RATIO OF (ENGINE DATA/DDA 8V71 DATA)



135% respectively, relative to the DDA diesel (See Figure 5).

CARBON MONOXIDE - The MAN methanol engine (catalyst equipped) was the best at 8% of the DDA for M100 and 16% for M85. The Iveco natural gas gave 68% of the DDA in its original state and this increased to 94% after efficiency and NO_x optimization efforts. The MAN diesel was next at 95%. The DDA yielded 1.9 g/bhp-hr, the Iveco diesel 105% of this and finally the Iveco propane gave a level of 111% (See Figure 6).

UNBURNED HYDROCARBONS - In this case, the methanol engine was also the best at 2% of the DDA for M100 and 11% for M85. The MAN diesel was next at 36% with the Iveco diesel registering 71%. The Iveco propane engine yielded 86%. The DDA at 1.4 g/bhp-hr and the Iveco natural gas at 114% of that were the largest emitters with the optimized version exceeding 200% of the 6V71 NA level (See Figure 7). It should be noted that the majority of the HC emissions of the natural gas engine is methane and may not be as reactive or damaging to humans or the environment as other hydrocarbons. In any case, they could be controlled by employing an exhaust oxidation catalyst as in the case of the MAN methanol engine.

NITROGEN OXIDES - The ADB transient cycle testing showed interesting results for emissions of nitrogen oxides. The lowest NOx levels were from the MAN methanol engine at 40% of that with the DDA 6V71 NA, when using M100. When M85 was used, the level increased to 64% owing to the presence of 15% (by volume) gasoline hydrocarbons. This was an indicator of the low NOx levels associated with methanol combustion since there was no reduction catalyst present.

The Iveco diesel gave 85% with the MAN diesel next at 89%. The DDA registered an average of 10.8 g/bhp-hr. The most interesting result was the high NOx levels of the two gaseous fuelled engines. The Iveco natural gas engine came in at 158% and the propane engine at 178% (See Figure 8). The optimized NGV version which employed lean mixtures at part load conditions resulted in an NOx value of 135% for the natural gas engine. This was a definite improvement over the original engine but still well above the DDA 6V71 NA. This indicates that an emission control system involving a reduction catalyst and appropriate air/fuel ratio control may be needed to meet future emission standards. Other emission controls such as Exhaust Gas Recirculation (EGR) under moderate engine load conditions may also be necessary to reduce NOx emissions to an acceptable level.

CONCLUSIONS

The MAN Methanol engine performed best overall with regards to the exhaust emissions examined. This is as expected since the engine was equipped with a conventional oxidation catalyst. It is a characteristic of methanol combustion that nitrogen oxide levels are relatively low. It was not within the scope of this program to evaluate exhaust aldehydes.

The diesel four-stroke engines exhibited higher cycle efficiencies than the DDA two-stroke.

All of the alternative-fuelled engines were extremely low in particulate emissions. The indication is that the alternate fuels can be useful in reducing the particulate levels of heavy-duty engines.

The engine torque curves indicate that comparable vehicle performance is available from each of the engines tested with some additional performance possible from the MAN methanol engine. LPG and NGV (gaseous fuels) also provide a fuel cost savings over diesel fuel in the current Ontario marketplace.

By operating the natural gas engine with a load-sensitive variable fuel mixture and spark advance, it was possible to obtain a substantial reduction (about 20%) in nitrogen oxide emissions, while slightly improving energy efficiency. The above gains were achieved with only a minor reduction in engine torque. To obtain the magnitude of nitrogen oxides reductions required to meet the proposed 1991 EPA bus standards, most likely a three-way catalyst/fuel mixture control system will be required.

With further engine/emission controls development, all of the alternative-fuelled engines tested in this program would be expected to meet the proposed 1991 EPA emission standards for urban buses.

ACKNOWLEDGMENTS

This program was funded by the Ontario Ministry of Energy and the Federal Energy, Mines and Resources Ministry of the Government of Canada.

REFERENCES

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APPENDIX 20

50272-101

REPORT DOCUMENTATION PAGE

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2.

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Advanced Natural Gas Fueled Engine-
Part 1: Design and Analyses

5. Report Date November 1986

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7. Author(s) R. Kamo, R. Walson, R.M. Kakwani and L. Kamo

8. Performing Organization Rept. No. FR8415-1186

9. Performing Organization Name and Address
Adiabatics, Inc.
630 South Mapleton
Columbus, Indiana 47201

10. Project/Task/Work Unit No.

11. Contract(C) or Grant(G) No.
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(G)

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8600 West Bryn Mawr Avenue
Chicago, Illinois 60631

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16. Abstract (Limit: 200 words)

The objective of this research program was to design an advanced natural gas engine (NG 1990) to be produced in the 1990's which will have high thermal efficiency and 40,000 hours durability of the valve train components before major engine overhaul. Preliminary design and feasibility of the NG 1990 advanced natural gas engine was completed. A natural gas engine simulation model predicts up to 43.6% brake thermal efficiency (5840 Btu/hp-hr BSFC) for the advanced engine with the advanced concepts like K-Miller cycle (early intake valve closing), lean burn combustion - A/F ratio = 24.5, high compression ratio up to 14:1, higher turbocharger efficiency of 63.2% overall, and axially stratified charge combustion system resulting in fast burning. The use of K-Miller cycle reduces the in-cylinder gas temperatures and allows engine operation at 14:1 compression ratio without knock tendencies. The design and analyses of the NG 1990 engine and its components like K-Miller system have been completed in this program.

17. Document Analysis a. Descriptors
Natural gas engine
Miller cycle
Natural gas engine simulation
Lean burn gas engine

b. Identifiers/Open-Ended Terms
Natural gas engine

c. COSATI Field/Group

18. Availability Statement
Release limited

19. Security Class (This Report)

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22. Price

(See ANSI–Z39.18) See Instructions on Reverse OPTIONAL FORM 272 (4–77) (Formerly NTIS–35) Department of Commerce

1.0 RESEARCH SUMMARY

Title	Advanced Natural Gas Fueled Engine- Part 1: Design and Analyses
Contractor	Adiabatics, Incorporated 630 South Mapleton Columbus, Indiana 47201
GRI Contract Number	5084-233-1048
Principal Investigators	R. Kamo, R. Walson, R. M. Kakwani, and L. Kamo
Report Period	April/1985-July/1986. Final Report
Objective	To design an advanced natural gas engine to be produced in the 1900's which will have less than 7000 Btu/hp-hr brake specific fuel consumption (up to 6000 Btu/hp-hr) and 40,000 hours durability of the valve train components before major engine overhaul.
Technical Perspective	The current natural gas engines in the 300 hp range are carburetted, spark ignited and have low compression ratios due to knock limitations. These factors affect the engine's brake thermal efficiency (32-35%) compared to 38-42% for diesel engines. In order to compete with the diesel engines, the future natural gas engines require technological changes and improved design as for the advanced diesel and gasoline engines. Thus, the more competitive advanced natural gas engines are needed in terms of cost, performance and durability.
Results	Preliminary design and feasibility of NG1990 advanced natural gas engines was completed. A natural gas engine simulation model predicts up to 43.6% brake thermal efficiency (5840 Btu/hp-hr) for the NG1990 engine with the advanced concepts. The design and analyses of the engine and its components like K-Miller system, etc. have been completed in the present program.

Technical Approach

The NG1990 engine will incorporate the advanced concepts like: K-Miller cycle (early intake valve closing), lean burn combustion, high compression ratio up to 14:1, higher turbocharger efficiency of 63.2% overall and axially stratified charge combustion system resulting in fast burning. These advanced concepts result in an efficient and durable natural gas engine. The use of the K-Miller cycle, that relies on the early intake valve closing, reduces the in-cylinder gas temperatures and allows engine operation at 14:1 compression ratio without knock tendencies. In addition, the axially stratified charge engine design with lean air to fuel mixtures will result in fast burning and is expected to produce lower emissions.

Project Implications

Natural gas fueled engines convert the heat energy of natural gas to shaft power. Therefore, these machines play a vital role in the use of natural gas in electric, and shaft power generation applications. The overall objective of the Gas Research Institute is to develop cost-effective, gas-fired heat-pump, cogeneration and power-cycle systems. Several such advanced systems developed in past years have reached the field testing and commercialization stage. Further, improvements of all these systems depend on the long range development of advanced, high efficiency, low cost, durable gas prime movers.

A technical development plan is mapped out in this report that could lead to an improved, high efficiency natural gas-fueled engine by 1990. The development scenario is well supported by technical data and by results from gas engine simulation computer calculations as well as by an extensive bibliography on gas engine technology. These results represent an important step in the direction of an advanced, modern gas engine.

2.0 INTRODUCTION

For years, natural gas fueled engines have remained nearly unchanged. Only recently, natural gas engine manufacturers expanded the gas engine's application from oil fields and sewage treatment facilities to prime power, peak sharing, and co-generation. However, with rising electricity and natural gas costs, it is apparent that the future natural gas engines should have higher performance and thermal efficiency. In addition, high reliability and durability, and longer life are required to make natural gas engines more attractive and cost effective to the natural gas customers. It is clear that the efficient and low maintenance natural gas engines will promote the use of natural gas as a fuel for the industrial engines.

The diesel engine industry has responded to the quest for higher engine performance by embarking on an adiabatic design with turbocompound cycle and a minimum friction engine concept. These concepts have been accelerated by the recent development of high temperature structural ceramic materials.

Recognizing the limited market for natural gas engines, the current gas engines are modified versions of their counterpart, the diesel engine. There appears little likelihood of designing and producing an engine optimized for the natural gas fuel. For example, the Cummins GTA 855 natural gas engine is a modified Cummins NH diesel engine. Also, the turbocharger on the GTA 855 natural gas engine is adapted from the Cummins' diesel engine However, the critical engine components, like valve train, piston rings and cylinder liner, in a natural gas engine can easily be modified from its diesel counterpart. This concept was also supported by our earlier work on "Formation of a Natural Gas Engine Data Base" for Gas Research Institute, Report Number GRI 85/0112 (1) *. It was also concluded that an advanced natural gas engine can be made more competitive with industrial diesel engines. This can be accomplished, firstly by improving the engine efficiency by incorporating the advanced concepts; and secondly, by increasing the engine durability by the use of wear-resistant ceramic parts. Adiabatics, Inc. along with Gas Research Institute can achieve these goals. This can be accomplished via the long range plan shown in Figure 2.0.1.

First, the existing carburetted (premixed) engine can be improved through performance and durability improvements. In the long term, a natural gas engine with high pressure fuel injection into the combustion chamber could be developed like diesel fueled engines. This development will also permit operation of an adiabatic natural gas engine with thermal barrier ceramic materials. The advances in natural gas fueled engines will run parallel to the advanced diesel engine, structurally, except for the valve train system. Thermodynamically, however, some differences between liquid fuel combustion and natural gas combustion exist.

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* Numbers in parentheses indicate references at the end of the report

The predicted progression of an advanced natural gas engine is shown in Figure 2.0.2 with energy comparisons for each engine design. The research program was divided into two parts: Part 1: NG1990 Engine Design and Analyses, and Part 2: Materials Investigation - Advanced Component Testing. The present report is about Part 1; a separate report is being submitted for Part 2.

2.1 Overall Project Objectives

The objective of the present result research program, "NG1990 Design and Analyses" was to investigate the design concepts for an advanced natural gas engine. These concepts would be applied to a commercial natural gas engine to provide higher engine efficiency, specific fuel consumption of at least 7000 Btu/hp-hr, and improved engine component durability- 40,000 hours for valve train before major engine overhaul. The advanced concepts considered in Part 1 are:

- Lean Combustion
- K-Miller Cycle (Early intake valve closing)
- Increased Turbocharger Efficiency
- Axially Stratified Charge Engine Design
- Alternate Multi-point Ignition Design.

2.2 Brief Project Description

The purpose of this program was to come forth with an advanced natural gas engine concept with performance that is competitive with the diesel engine, and with improved durability far beyond the current levels. In order to reach the goals of a minimum specific fuel consumption (7000 Btu/hp-hr, 36.4% brake thermal efficiency minimum) and 40,000 hours durability before major overhaul, the following aspects of engine design were investigated:

- Lean burn combustion utilizing axial stratified charge for fast burning of the natural gas.

- Miller cycle (early intake valve closing) to reduce in-cylinder gas temperatures and reduce combustion chamber wall temperatures in order to permit engine operation at higher compression ratios without knocking.

- A high efficiency turbocharger able to operate at higher boost pressures.

- Alternate designs using either dual admission valves with a single spark, or multi-point spark gap ignition points within the cylinder.

- Advanced metal and ceramic components for valve guides, valve seat inserts, valves, cylinder liners and piston rings to extend durability to 40,000 hours.

The work plan for the program is presented in Section 3.1.

2.3 Rationale for Undertaking the Project

This project was being pursued as part of GRI's effort to apply existing and new technologies to the industrial natural gas sector in order to make natural gas fueled engines more competitive with diesel fueled engines. Since the benefits of using insulated diesel engines have been well established, the transfer of this technology to manufacturers of natural gas industrial engines and component suppliers will be systematic.

The advanced concepts for the natural gas engines investigated in the present work will aid in improving the efficiency and durability of gas engines. This will result in an expanded market share of natural gas engines, which will lead to a higher level of customer acceptance and increased natural gas consumption by customers. The long range benefits of this program to natural gas producers, to natural gas users, and to GRI are expected to be substantial.

2.4 Benefits to Natural Gas Customers

The benefits to natural gas customers will be significant. The technical advantages of a diesel-competitive, natural gas industrial engine will be in terms of increased performance and extended durability (life). This report shows the performance of the natural gas engine (in this case, Cummins GTA 855 gas engine) can be improved by incorporating the K-Miller cycle, lean burn combustion, an efficient turbocharger, and a fast burn design. Also, the durability of natural gas engines can be extended by using wear-resistant ceramic materials for the valve train, cylinder liner, and piston rings components.

The maintenance costs of the advanced natural gas engine can subsequently be reduced. Since the durability of the engine will improve, the major teardown time interval can be increased from 12,000 to 36,000 hours. This will lower the number of service personnel as well as lower the inventory of replacement parts. These improvements have been estimated to reduce the potential maintenance costs of an industrial size engine for 52,560 hours of continuous duty from $2,648 to $1,148 per year when averaged over a six year period (1). This can result in a significant cost savings to natural gas customers, along with the reduced fuel consumption due to increased engine efficiency.

APPENDIX 21

FEV of America **UPDATE**

Volume 2 Issue 2 November 1988

FILE
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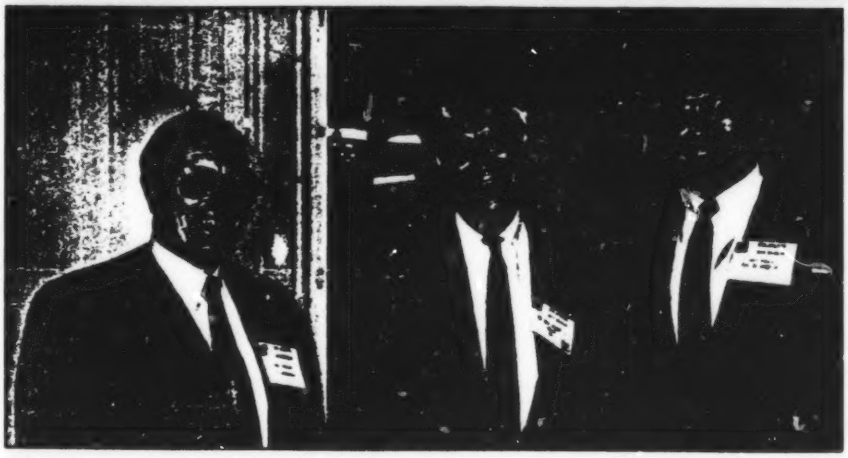


FEV Motorentechnik President Prof. Dr. Franz Pischinger, FISITA President Dr. Heinz W. Hahn and FEV of America President Gary W. Rogers exchange greetings at the recent FISITA Congress held in Dearborn.

FISITA '88 Draws International Delegation to Dearborn

For the first time in its 40-year history, the Congress of the Federation Internationale des Sociétiés d'Ingénieurs des Techniques de l'Automobile (FISITA) was held in North America, September 25-29 in Dearborn, Michigan and September 30 in Washington, D.C. The XXII FISITA Conference was hosted by the Society of Automotive Engineers, Inc. (SAE).

This International Federation of Automobile Engineering Societies is comprised of 22 associated engineering organizations worldwide.

Dr. Heinz W. Hahn, FISITA President, offered a welcoming address, with the official opening provided by Conference Honorary Chairman, Roger B. Smith, Chairman and Chief Executive Officer, General Motors Corporation.

Keynote addresses concerning the conference theme "Automotive Systems Technology - The Future," were offered by Roger Smith, Raymond H. Levy, Chairman and Chief Executive Officer, Regie Nationale des Usines Renault (read by John B. Curcio, Chairman and Chief Executive Officer, Mack Trucks, Inc.) and Tadashi Kume, President and Chief Executive Officer, Honda Motor Co., Ltd., providing North American, European and Far Eastern global perspectives.

One very important issue presented during the Keynote addresses was Renault Chairman Levy's estimate that the number of employees in the European automotive industry will be reduced by 50% compared with 1986 figures, yet production will remain at about the same level. Levy stated that he believes the auto industry will reduce staff by placing more responsibility on manufacturing and production personnel, introducing additional computer technology which will replace certain administrative jobs which, today, primarily obtain and sort data, displacing sub-assembly from current production lines and increasing the use of outside suppliers for design and production.

Continued on Page 4

'88 SAE FTTC Sets Attendance Record

The recent 1988 SAE Conference & Exposition on Future Transportation Technology (FTTC) was one of the most successful such meetings ever held. Nearly 620 people from throughout the country attended the event held in San Francisco August 7-11, exceeding attendance of the previous FTTC in Seattle by more than 30%.

According to SAE Staff Engineer Kenneth Kaufmann, the technical program, organized by Gary Rogers, FEV of America

Continued on page 2

FEV/Navistar International Present Paper At Alcohol Fuels Symposium

A featured attraction at the VIII International Symposium on Alcohol Fuels, November 13-16, in Tokyo is the presentation of a paper co-authored by Navistar International Transportation Corporation and FEV. Dr. Ernst Scheid, FEV Motorentechnik department manager, will present the paper, "Conversion of a Navistar DT-446 Diesel Engine to Methanol Operation."

The paper outlines the development of a methanol engine, based upon the Navistar DT-466 diesel engine. In this particular approach, a hot surface ignition (glow plug) system is used to assist in methanol ignition.

Those interested in receiving a copy of the paper from the proceedings can contact Bob Last, FEV of America.

The FTTC banquet featured a presentation by Mr. Gary Dickinson, GM Vice President and Group Executive. Some of the distinguished guests joining Gary Rogers at the banquet head table included (l-r) Dr. Robert Donoho, General Chairman; George Aravosis, SAE President; Keith Parmee, Chairman, Auto. Div. IMechE; Dr. Charles E. Wilkes, VP Technology, GenCorp Automotive.

FTTC continued ...

President, "was certainly the major reason for the attendance record, the highest in the last eight years for the (FTTC) meeting and the former West Coast Meeting. That attendance was only exceeded five times in the last 20 years."

Kaufmann added that the "selection of session topics and the organizers' dedicated close guidance and attention to deadlines were a very major reason for the success, which was also attested to by the session attendance and the many favorable comments received about the technical program."

Concerning the technical program, Rogers stated "I was very pleased with the quality of the papers presented, and all authors and reviewers should be commended. A testament to the quality and popularity of the papers is the number which were purchased during the conference. According to SAE, paper sales for this year's FTTC were almost 2.5 times the number sold last year."

The '89 SAE FTTC, to be held in Vancouver, B.C., will include many of the successful topics featured in this year's conference and should continue to promote the growing interest in the Future Transportation Technology theme.

Gary Rogers Testifies at Energy Commission Hearing

Gary W. Rogers, FEV of America president, recently testified in the California Energy Commission's (CEC) Hearing on "Draft Energy Development Report, Draft Energy Technology Status Report and Notice of Availability." In particular, Rogers was asked to present testimony concerning specific areas of advanced engine technology which are of interest to the CEC in its goal to reduce exhaust emissions, expand the use of alternate fuels and improve the overall fleet fuel economy to reduce the consumption of petroleum derived fuels.

Rogers' testimony focused on future developments in alternative fuels and energy efficiency measures for transportation technology. He discussed the advancements which FEV is achieving in technologies such as alcohol fuels (methanol, ethanol, ignition improvers), flexible fuel engines, compressed natural gas (CNG), hydrogen fuels, liquefied petroleum gas (LPG), propane, butane, biogas and exhaust gas aftertreatment (catalysts, particulate traps). Some interesting points concerning future development areas are summarized below:

Flexible Fuel Vehicles

Advancements in the accuracy and reliability of fuel composition sensor (to determine the amount of methanol/gasoline in the mixture).

Development of specialized catalysts for reduction of unburned hydrocarbon (unburned fuel) and aldehyde emission reduction, in addition to a requirement for improvements in high mileage control of catalyst conversion efficiencies.

High Speed DI Methanol Engines

- 30% improvement in fuel economy over SI methanol engines
- Reduced carbon dioxide emission (70% of SI methanol engines)
- Reduced aldehyde emissions (compared to SI methanol engines)
- Simple oxidation catalyst

Those wishing to receive copies of the prepared testimony exhibits should contact Sue Kelly at FEV of America (313/352-1400).

Heavy-Duty Diesel Cycle SI CNG Engines

Combustion Concept	Maximum[1] Power (Percent)	BSEC[1] (Percent)	Exhaust Emissions[2] CO (Percent)	HC	NOx
Naturally Aspirated					
1. Combination Fuel Control & Throttling Ignition Timing for Best Fuel Economy	100	100	Not Crit.	160	250
2. Combination Fuel Control & Throttling Ignition Timing 10°CA Retarded	95	105	Not Crit.	100	160
3. Full Load: λ = 1.0 + EGR Part Load: λ = 1.4 + Throttling Ingition Timing 10°CA Retarded	84	100	Not Crit.	130	90
4. λ = 1 Concept, 3-Way-Catalyst	100	103	Not Crit	60	low
Turbocharged					
5. Lean Burn (Homogeneous Charge) Oxidation Catalyst	116	98	Not Crit	low	20-70[3]
6. Inert Gas Charge Stratification (Estimated Performance)	110+	100	Not Crit.	low	low

(1) Values Normalized to NA BSEC Case 1.
(2) Values Normalized to ECE 13 Mode Standards
(3) Charge Air Temperature 50 - 100°C

FEV's Perspective

Study Shows Better Performance from Long Stroke Engines

During the XXII FISITA Congress held in Detroit on September 25-30, Dr. Peter Kreuter, FEV's Manager of Engine Design and Development, Dr. Bernhard Gand, a supervisor in the Design and Development Department, and Prof. Pischinger presented a paper on "The Effects of the Combustion Chamber Shape on the Combustion Process of S.I. Engines for Different Stroke to Bore Ratios."

FEV undertook an extensive internal research project to describe the influence of the stroke to bore ratio on the combustion process when different combustion chamber shapes, compression ratios, and/or cylinder displacements are used.

A modular, single-cylinder test engine featuring interchangeable components was employed, allowing the entire range of today's motor vehicle engines to be represented by the study. The criteria used to evaluate the resulting engine variants were internal efficiency, exhaust emissions, indicated mean effective pressure (IMEP), standard deviation of the IMEP (an indicator of operational quality), and power density.

Two combustion chamber types were investigated, disc-shaped and squish (Heron) type.

The hydrocarbon emissions, for a constant air/fuel ratio, were higher for the squish type combustion chamber than for the disc type with the exception of very lean mixtures (Lambda > 1.1). However, use of the squish chamber resulted in better operational quality. For a given combustion chamber shape the HC emissions with the long stroke variant were about 25% better (Lambda = 1.0) than the short stroke engine. In the region of air/fuel ratio between 0.9 and 1.2, the influence of stroke to bore ratio on HC emissions was predominant.

For a given combustion chamber type, the internal efficiency vs. NOx characteristics of the long stroke variant were superior.

The fuel consumption values are indicated in the accompanying figure. With a given bore to stroke ratio, the squish chamber consistently yielded better fuel consumption than the disc chamber. However, variation in the bore to stroke ratio was found to have a very pronounced effect on the fuel consumption values. The relative ISFC for the squish chamber was found to be more than 16% lower for the long stroke variant, without degradation in the operational quality, in comparison to the short stroke engine. Similar results were obtained with the disc chamber.

The second parameteric variation to be considered was the compression ratio. Lean burn curves were obtained for the squish and the disk chamber at two compression ratios (10:1 and 18:1).

The results indicated that, especially for the squish chamber, there is a sharp increase in efficiency, without degradation in the operational quality, as the air/fuel ratio becomes increasingly lean.

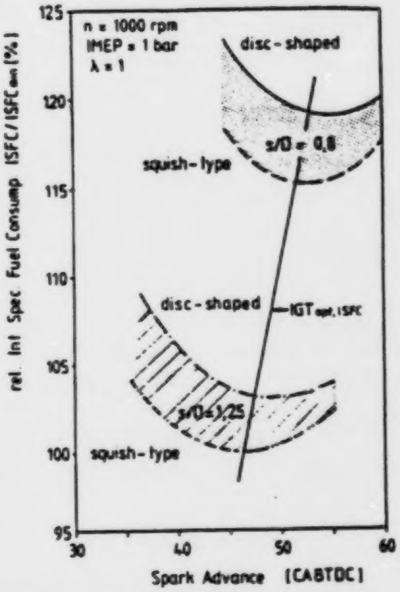
The HC emissions were obtained and evaluated as a function of the operational quality with data taken at both compression ratios. The results indicate that increasing the compression ratio results in a corresponding increase in the HC emissions with the ignition timing optimized for best fuel consumption. Consequently, to utilize the positive potential benefits of increased compression ratio, careful design of the combustion chamber and a strategy for late injection timing is necessary.

At lower IMEP levels, higher compression ratios provide better efficiency. However, the highest IMEP was obtained with a compression ratio of 10:1. If consideration was given to variable compression ratio as a function of load, significant improvements in fuel consumption (up to 7%) would be possible up to about 70% load.

The final parameter studied was the influence of varying the cylinder displacement. Experiments were conducted with data obtained for two different cylinder displacements (300 cm3 and 450 cm3) at both stroke to bore variants.

In general, performance of the long stroke engine was better, up to medium speed. At higher engine speeds, the power density of the short stroke engine is better. However, the short stroke engine, for either swept volume, showed higher fuel consumption values across the entire speed range. The influence of the stroke to bore ratio was observed to be considerably larger than the influence of the swept volumes at low speed. Conversely, at higher speeds, the influence of the swept volume was more significant.

A critical evaluation of the results indicates the long stroke engine has unique advantages in terms of higher efficiencies, better operational quality, and improved emissions and fuel consumption. In comparison, the short stroke engine exhibits higher power density. An increase in compression ratio and use of the squish type combustion chamber provide significant improvements in efficiency and operational quality at part load.



Indicated Specific Fuel Consumption Near Idle Operation

A written presentation of the experimental work described above is available by contacting FEV of America.

Industry News

California Appropriates $60 Million for Clean Fuel School Bus Program

California Governor George Deukmejian has signed into law legislation which appropriates $60 million for the Safe School Bus Clean Fuel Efficiency Demonstration Program. The funds, which are collected and distributed by the Federal Government, are part of the Petroleum Violation Escrow Account (PVEA), created by fines paid by oil companies for overcharging customers in the 1970's.

In addition to the $60 million, the legislation directs the appropriation of additional program funding as future PVEA funds become available. The total funding package could exceed $100 million.

The overall objective of the program is to demonstrate direct energy savings (without adverse air quality effects), reduce the waste of tax money spent due to inefficient energy use, and promote the use of advanced energy products. A particular goal of the program is to allow the California school districts to upgrade their fleets in a cost effective, energy efficient and environmentally sound manner. To aid in achieving this goal, the legislation requires that at least 35 percent of the replacement school buses be operated on methanol or other low-emission, clean-burning fuels.

According to the California Energy Commission (CEC), the state has nearly 12,000 publicly-owned school buses, over half of which were manufactured before the federal government imposed minimum safety standards in 1977. These buses exhibit high repair costs, produce excessive exhaust emissions, are not fuel efficient and lack desirable safety standards.

The California Highway Patrol, in cooperation with the California Department of Education, will conduct engineering and economic analyses of minimum standards for school bus replacement, reconditioning, inspection and preventative maintenance. The CEC will rely on input from the Highway Patrol and Education Department in determining criteria for bus replacement.

For more information concerning this very important appropriation, contact the School Bus Demonstration Program, Transportation Technology Fuels Office, California Energy Commission (916/324-3527).

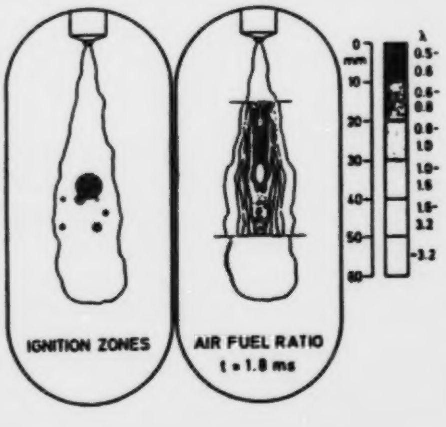
FISITA continued ...

The issue of reduction in the automotive work force was also discussed by Dr. David E. Cole, Director of the Transportation Research Institute at the University of Michigan in his presentation entitled "The Automotive Industry in Transition," during a session addressing the social/economic impact of advancing technology. Dr. Cole likewise predicts a significant reduction in the U.S. automotive work force due to over capacity and the need to become more cost effective to compete on a global basis.

During the FISITA conference, a special CIMAC session was held concerning research, development and production of diesel engines. Prof. Franz Pischinger, FEV Motorentechnik President, participated in this special meeting with his presentation "Influence of Fuel, Injection and Combustion Chamber Wall on Self Ignition of Diesel Sprays."

The results of the investigation indicate that the ignition is not only influenced by fuel quality and the state of the air, but also by the injection equipment characteristics and the volume of the fuel injected. It was also observed that cetane number is not itself sufficient to define the ignition performance of fuels, based upon investigations with fuels exhibiting different physical properties and cetane numbers. Empirical results using high speed Schlieren photography and spontaneous Raman Spectroscopy were presented which showed that the self ignition process seems to occur at locations where a rich air/fuel mixture, between 0.6 and 0.8, was identified.

Although FISITA requested an oral presentation by Prof. Pischinger, FEV has prepared a technical publication concerning these important investigations. Copies may be obtained by contacting Bob Last at FEV of America (313/352-1400).



Relationship between local air/fuel ratio and ignition (n-Heptane fuel, injection quantity 14 mm3/stroke, injection hole diameter 0.2 mm, air pressure 45 bar, air temperature 500 degrees C)